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(71) Applicant (for all designated States except US): KELSEY-HAYES CO. [US/US]; 12000 Tech Center Drive, Livonia, MI 48150 (US).

(72) Inventor; and
(75) Inventor/Applicant (for US only): GANZEL, Blaise, J. [US/US]; 3317 Alton Court, Ann Arbor, MI 48105 (US).

(74) Agent: BLAKE, Scott, A.; MacMillan, Sobanski & Todd, LLC, 4th floor, One Maritime Plaza, 720 Water Street, Toledo, OH 43604 (US).

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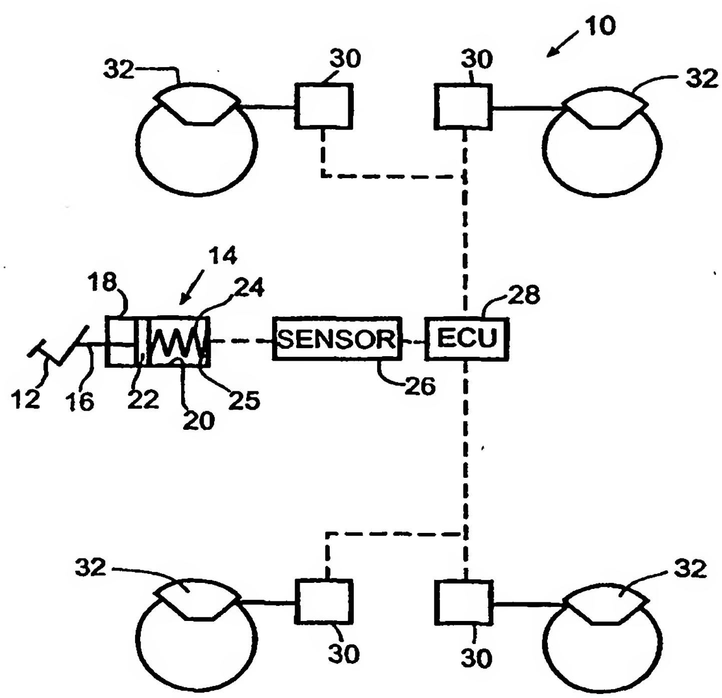
(54) Title: PEDAL SIMULATOR SPRING FOR VEHICLE BRAKE SYSTEM

(57) Abstract

A spring (24) which may suitably be used in a pedal simulator of a brake system to provide a progressive non-linear spring rate. In a first embodiment, the spring is formed from a wire having a varying cross-sectional area along the length of the spring such that the wire is cylindrically wound into a helical coil compression spring having a generally constant pitch (L). In a second embodiment, the spring is formed as a wave spring. Methods of forming the spring are disclosed, including a method of forming the spring by removing material from the outside diameter of the spring to form a non-cylindrical outer contour. The use of the spring in a pedal simulator (14) of an advanced brake system is also disclosed.



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TITLE

PEDAL SIMULATOR SPRING FOR VEHICLE BRAKE SYSTEM

CROSS REFERENCE TO RELATED APPLICATIONS

This application claims the benefit of U.S. Provisional Application No. 60/069,084 filed December 10, 1998.

BACKGROUND OF THE INVENTION

This invention relates in general to springs, and in particular to compliance springs having a non-linear spring rate for use in a brake system pedal simulator.

In conventional hydraulic vehicle brake systems, the force used to pressurize hydraulic fluid to operate the vehicle brakes comes from the vehicle operator pressing on the vehicle brake pedal to actuate the master cylinder of the brake system. Normally, the resultant movement of the brake pedal linkage to the master cylinder is also used to actuate a vacuum or hydraulic boost system to provide an assisting force which aids the force provided by the vehicle operator in actuating the master cylinder and thus operate the vehicle brakes.

In certain recent advanced vehicle brake systems, such as so-called brake-by-wire brake systems, it is known to supply all of the force to operate the vehicle brakes from mechanical or electrical devices such as pumps or linear actuators. Sensors provide control signals used to control the operation of these devices supplying the force for operating the vehicle brakes. The sensors can measure the amount of movement of the brake pedal, the force with which the operator steps on the brake pedal, or both to generate the control signal. Since the brake pedal in these advanced systems does not directly actuate the vehicle brakes, it is common to provide a pedal simulator (also known as a compliance unit) to provide a simulated load on the brake pedal which provides a reaction force to the operation

of the brake pedal similar to that experienced during the operator of a brake pedal in a conventional brake system. By providing a simulated load on the brake pedal similar to that experienced during the operation of a brake pedal in a conventional brake system, the transition from conventional brake systems to advanced brake systems having a pedal simulator is eased, since the driver of a vehicle with such an advanced brake system will enjoy a "normal" or similar pedal feel. An additional purpose of the use of a pedal simulator is to provide pedal displacement or travel during operation of the brake pedal.

A plot of pedal force (the force exerted by the operator on the brake pedal) versus pedal travel in a conventional brake system is typically non-linear. Typically, conventional brake systems experience a relatively slow rate of increase in pedal force during an initial portion of brake pedal travel from a brake released position to a partially applied brake position, and a relatively high rate of increase in pedal force during a final portion of brake pedal travel from a partially applied brake position to a maximum effort (full applied) brake position, i.e., a progressive rate application of load on the brake pedal.

In the past, pedal simulators have included helical coil springs formed into a conical shape to provide a progressive non-linear response, such as those disclosed in International Patent Application No. PCT/US98/02613. However, the conically shaped coil springs are generally relatively large or long in length due to the relatively large number of coils required since a selected number of coils will bottom out or contact adjacent coils at relatively low stress levels. Thus, the pedal simulator housing must be sized to accommodate the relatively large sized spring which can also add to the overall weight and cost of the pedal simulator.

Nested helical coil spring arrangements have also been used in the past, such as those disclosed in International Patent Application No. PCT/US98/02613. The nested helical coil spring arrangements include a plurality of constant pitch helical coil springs having different linear spring rates or spring coefficients. The

springs can be arranged in a nested manner such that they are compressed in a parallel manner. Although the nested spring arrangement provides a progressive non-linear response, it is generally more costly due to the plurality of springs.

SUMMARY OF THE INVENTION

This invention relates to a spring which may suitably be used in a pedal simulator of a brake system to provide a progressive non-linear spring rate. In a first embodiment, the spring is formed from a wire having a varying cross-sectional area along the length of the spring such that the wire is cylindrically wound into a helical coil compression spring having a generally constant pitch. In a second embodiment, the spring is formed as a wave spring. Methods of forming the spring are disclosed, including a method of forming the spring by removing material from the outside diameter of the spring to form a non-cylindrical outer contour. The use of the spring in a pedal simulator of an advanced brake system is also disclosed.

Various objects and advantages of this invention will become apparent to those skilled in the art from the following detailed description of the preferred embodiment, when read in light of the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a schematic diagram of a first embodiment of a brake system having a pedal simulator using a spring having a non-linear spring rate, in accordance with the present invention.

Fig. 2 is a schematic plot of pedal travel versus pedal force for the spring of the pedal simulator of Fig. 1.

Fig. 3 is a sectional view of a first embodiment of a spring, in accordance with the present invention, which can be used for the spring of the pedal simulator of Fig. 1.

Fig. 4 is a sectional view of a second embodiment of a spring, in accordance with the present invention, which can be used for the spring of the pedal simulator of Fig. 1.

Fig. 5 is an elevational side view of a third embodiment of a spring, in accordance with the present invention, which can be used for the spring of the pedal simulator of Fig. 1.

Fig. 6 is a front elevational view of the spring of Fig. 5.

Fig. 7 is an elevational side view of a fourth embodiment of a spring, in accordance with the present invention, which can be used for the spring of the pedal simulator of Fig. 1.

Fig. 8 is an elevational side view of a fifth embodiment of a spring, in accordance with the present invention, which can be used for the spring of the pedal simulator of Fig. 1.

Fig. 9 is an elevational side view of a sixth embodiment of a spring, in accordance with the present invention, which can be used for the spring of the pedal simulator of Fig. 1.

Fig. 10 is a schematic diagram of a second embodiment of a brake system having a pedal simulator using a spring having a non-linear spring rate, in accordance with the present invention.

Fig. 11 is a schematic diagram of a third embodiment of a brake system having a boost valve actuated by a spring of a pedal simulator having a non-linear spring rate, in accordance with the present invention,

Fig. 12 is a cross-sectional view of an alternate embodiment of the base brake valve of Fig. 11.

Fig. 13 is a cross-sectional view of an alternate embodiment of the pedal travel simulator valve of Fig. 12.

Fig. 14 is a cross-sectional view of an alternate embodiment of the boost valve and the pedal travel simulator of Fig. 13.

Fig. 15 is an enlarged sectional view of the boost valve of Fig. 14.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring now to the drawings, there is illustrated in Fig. 1 a simplified schematic of a first embodiment of a vehicular brake system, indicated generally at 10, which is commonly referred to as a brake-by-wire system. The brake system 10 includes a brake pedal 12 which is operable by the driver of the vehicle. The brake pedal 12 is operatively connected to a pedal simulator, indicated generally at 14, by an actuating arm 16. The pedal simulator 14 provides a simulated load on the brake pedal 12 to provide a reaction force to the brake pedal 12. The pedal simulator 14 can be configured to provide a reaction force similar to a conventional brake system or can be configured to provide other desired reaction forces. The pedal simulator 14 includes a housing 18 having a bore 20 formed therein. A piston 22 is slidably disposed in the bore 20. The actuating arm 16 is operatively coupled to the piston 22. A spring 24 is interposed between the piston 22 and an end 25 of the housing 18. The spring 24 biases the piston 22 against the actuating arm 16. As will be explained in detail below, the spring 24 can be any suitable spring arrangement having a non-linear spring rate, and more preferably having a non-linear progressive spring rate.

The brake system 10 further includes an input sensor 26. The input sensor 26 senses the desired braking input of the driver through operation of the brake pedal 12. The input sensor 26 can be any suitable sensor capable of generating an input signal relative to the driver's desired braking input. For example, the input sensor 26 may sense displacement of the brake pedal 12 or any linkage associated therewith, such as the actuating arm 16. Alternatively, the input sensor 12 may sense the input force applied by the driver of the vehicle, such as for example, by a load cell which is incorporated into the brake

apply force path. Of course, these examples of an input sensor 26 are not intended to be limiting, and other driver sensors may be suitable for use as an input sensor 26.

The input signal generated by the input sensor 26 is transmitted to an electronic control unit (ECU) 28. The ECU 28 is electrically connected to brake actuators 30 which actuate associated wheel brakes 32. The brake actuators 30 and wheels brakes 32 can be any suitable mechanism for applying a braking force to the associated wheel of the vehicle. For example, the brake actuator 30 can be a linear actuator operable by an electric motor (not shown) which actuate brake pads (not shown) associated with the wheel brake 32. Alternatively, the brake actuator 30 can be a hydraulic power cylinder. Based on the input signal from the input sensor 26, the ECU 28 transmits a control signal to the brake actuators 30 to actuate the associated wheel brakes 32 in accordance with the driver's desired braking input. The ECU 28 may also be connected to other sensors (not shown) which sense various vehicle parameters, such as vehicle and wheel speeds. Thus, the ECU 28 can control the brake actuators 30 based on signals other than the input signal, for example, for controlling the brake actuators during an anti-lock or traction control braking operation.

As stated above, the spring 24 preferably has a non-linear spring rate and more preferably a non-linear progressive spring rate as graphically represented in the schematic plot of Fig. 2. The plot generally illustrates the pedal force, e.g., the force exerted by the operator on the brake pedal 12, versus pedal travel for a preferred embodiment of a spring 24. Note that the spring rate is a progressive non-linear rate such that there is a relatively slow rate of increase in pedal force during an initial portion of brake pedal travel from a brake released position to a partially applied brake position, indicated generally at "A", and a relatively high rate of increase in pedal force during a final portion of brake pedal

travel from a partially applied brake position to a maximum effort (full applied) brake position, indicated generally at "B".

There is illustrated in Fig. 3, a first embodiment of a spring 40, in accordance with the present invention, having a progressive non-linear spring rate characteristic of the spring rate shown in the plot of Fig. 2. The spring 40 can be used in place of the spring 24 of the brake system of Fig. 1. The spring 40 is a cylindrically wound helical coil compression spring. "Cylindrically wound", as used herein, means that the inner contour of the helix formed by the spring 40 has a constant inner diameter, indicated generally at "I", along an axis "X", as if the spring were wound around a cylinder. As shown in Fig. 3, the coils of the spring 40 vary in cross-sectional area along the length of the spring 40. Note that the spring 40 does not have an outer contour with a constant outer diameter "D" along the axial length of the spring 40. The outer contour of the spring 40 may thus be termed "non-cylindrical" or "axially variant", such that the outer contour varies as a function of displacement along the axis of the spring 40.

Preferably, the pitch or axial distance "L" between adjacent coils is generally constant. This constant pitch is formed by maintaining a constant helical angle when the spring 40 is wound. However, the spring 40 can have ends 42 which are typically closed, i.e., permanently bent to abut against the adjacent coil such that the coils at the ends 42 do not significantly affect the spring characteristic of the spring 40 as a whole. As used in this application, when it is said that a spring has a constant pitch or constant helical angle, it should be understood that this term is meant to include springs having a constant helical angle except where the end coil is bent closed. The ends 42 of the spring 40 can also be ground flat, thereby providing generally flat contact surfaces perpendicular to the axis X, such as for engagement with the piston 22 and the end 25 of the housing 18 of the pedal simulator 14 illustrated in Fig. 1.

As indicated above, the outer contour of the spring 40 does not have a constant diameter. The spring 40 illustrated in Fig. 3 has a generally conical outer contour, i.e., having a decreasing outer diameter D from the left-hand end of the spring 40 to the right-hand end of the spring 40, as viewed in Fig. 3. Thus, the cross sectional area of the wire forming the spring 40 decreases from the left-hand end of the spring 40 to the right-hand end of the spring 40, as viewed in Fig. 3. This gives the spring 40 a progressive non-linear spring rate characteristic. The spring 40 is initially relatively easy to compress from the fully expanded, uncompressed state, due to the relatively small cross sectional area at the right-hand end of the spring 40, and the relatively large number of active spring coils. As the spring 40 is compressed, such as by movement of the piston 22 if the spring 40 is used for the spring 24 of the pedal simulator 14 of Fig. 1, the right-hand end coils of the spring 40 twist sufficiently that adjacent turns of the coil begin to abut against one another, a condition commonly referred to as "going solid". Once a turn of the spring 40 has gone solid, the solid portion of the spring 40 no longer acts as a spring. A decreasing number of active coils as the spring 40 goes solid results in a progressive or increasing spring rate of the spring 40. Additionally, the remaining portion of the spring 40 (which has not gone solid) has coils with relatively larger cross sectional areas, further contributing to a progressive spring rate as the spring 40 goes solid.

The spring 40 can be formed by any suitable method. One method for forming the spring 40 is to first form a helical coil compression spring from wire having a generally constant pitch such that the distance between the adjacent coils is generally constant. Of course, the ends of the spring can be closed and ground flat. The desired outer contour of the spring is then formed. This may be accomplished, for example, by turning the spring 40 on a lathe, or grinding the spring 40 with a grinding wheel. Alternatively, the inner contour of the spring 40 may formed to provide the varying cross-sectional shape. For the spring 40

illustrated in Fig. 3, the right-hand portion of the spring 40 is ground down more than the left-hand portion, thereby forming the conical shaped outer contour. Although the wire used can have any cross-sectional shape, it is believed that the use of wire with a rectangular cross sectional shape provides stable geometry when the spring goes solid and adjacent faces of the coil are in contact with one another. As a third step, if required, the spring is heat treated and otherwise finished as desired, for example, by, quenching, tempering, coating, dying, plating or other similar processes useful in forming springs, as dictated by the application in which the spring 40 is to be used. Note that the second step of forming the outer or inner contour may be performed

Another method for forming the spring 40 is to first provide a length of wire having a non-uniform cross-section, for example, a wire having a decreasing cross-sectional area along the length of the wire. The wire is then wound to form a helical compression spring, thereby forming a spring having a non-linear spring rate. The wire can be cylindrical wound or wound to any other suitable shape, such as into a conical helical shape. Of course, the ends of the spring can be bent closed and ground flat. The spring would exhibit a non-linear spring rate as the spring is compressed such that the coils having the smaller cross-sectional area will axially abut one another as the spring goes solid. It will be appreciated that a spring formed according to this alternate embodiment may have a cylindrical outer contour and a non-cylindrical inner contour because of the axial variation in the cross-sectional area of the wire forming the spring. Such an effect may also be achieved by machining the inner contour of a conventional helical coil compression spring. It will also be appreciated that a spring formed according this alternate embodiment may instead have both a cylindrical outer contour and cylindrical inner contour, but have a non-constant distance between adjacent coils of the spring because of the axial variation in the cross-sectional area of the wire forming the spring, even though a centerline of

the wire forming the spring exhibits a constant helical angle. It is also contemplated that a spring according to the invention may be formed with any combination of non-cylindrical outer contour, non-cylindrical inner contour, or non-constant distance between sequential adjacent coils of the spring.

The spring 24 of the pedal simulator 14 of Fig. 1 can have other various contours of the outer diameter to achieve desired spring characteristics. For example, there is illustrated in Fig. 4 a second embodiment of a spring 50, in accordance with the present invention, having a progressive non-linear spring rate characteristic of the spring rate shown in the plot of Fig. 2. The spring 50 is a helical coil compression spring having an "hourglass" outer contour such that a central portion 52 of the spring 50 has a smaller diameter than outer portions 54 of the spring 50. The spring 50 can be formed by first providing a cylindrically wound spring formed from a wire having a constant cross-section, and then grinding down the central and outer portions 52 and 54 until the spring 50 has the desired hourglass outer contour shape.

There is illustrated in Figs. 5 and 6 a third embodiment of a spring 60, in accordance with the present invention, having a progressive non-linear spring rate characteristic of the spring rate shown in the plot of Fig. 2. The spring 60 is a wave spring, also known as a wave washer spring. The spring 60 has a circular band 61, formed about an axis "X", having a plurality of axially extending oscillating bends formed thereabout. In the illustrated embodiment of the spring 60 shown in Figs. 5 and 6, the spring 60 has six bends forming three convex bends 62 and three concave bends 64. Note that the terms "convex" and "concave" as used herein are relative terms used to reference the various orientations of the bends of the spring 60 as viewing the illustrations and are not meant to be limiting. Of course, the spring 60 can have any suitable number of bends. Note that the band 61 has a pair of ends 66 and 68 which overlap one

another. If desired, the spring 60 can be configured such that the ends 66 and 68 do not overlap each other, but instead form a gap therebetween.

The spring 60 can be used for the spring 24 of the brake system 10 by positioning the concave bends 64 against the piston 22 and the convex bends 62 against the end 25 of the housing 18. The axial distance between the convex bends 62 and the concave bends 64 can be formed at any suitable distance to provide a desired spring rate. Other characteristics of the spring 60 which can be changed to affect the spring rate are, for example, the material used for the band 61, the thickness of the band 61, and the radial width of the band 61. Although the band 61 is shown and described as circular, the band 61 can have other shapes, such as square or linear.

The spring 60 can also be configured such that the spring 60 includes multiple wrappings or coils. For example, there is illustrated in Fig. 7, a fourth embodiment of a spring 70, in accordance with the present invention. The spring 70 is a wave spring similar to the wave spring 60 as described above, except having a pair of coils 72 and 74 which abut one another. The coils 72 and 74 have a plurality of convex bends 76 and a plurality of concave bends 78 which are positioned adjacent one another in a "nested" or "parallel stacked" arrangement. Thus, the radially extending adjacent surface areas of the coils 72 and 74 facing one another are substantially in direct contact with one another. Generally, multiple coils in a nested arrangement proportionally increases the spring rate of the spring 70 relative to the number or coils, thereby providing a relatively high spring rate for a relatively short deflection distance.

There is illustrated in Fig. 8 a fifth embodiment of a spring 80. The spring 80 is a wave spring having a plurality of coils 82, 84, and 86. Unlike the spring 70 of Fig. 7, the coils 82, 84, and 86 of the spring 80 have a plurality of convex bends 88 and a plurality of concave bends 89 which are not positioned in a nested arrangement. Instead, the convex bends 88 are positioned adjacent

concave bends 89 of adjacent coils 82, 84, and 86, which is commonly referred to as a "crest-to-crest" or "series stacked" arrangement. Generally, multiple coils in a crest-to-crest arrangement proportionally decreases the spring rate of the spring 80 relative to the number of coils, thereby providing a relatively precise force versus deflection characteristic.

There is illustrated in Fig. 9 a sixth embodiment of a spring 90, in accordance with the present invention. The spring 90 is a wave spring having a plurality of coils 92, 93, 94, and 95 which are positioned in a crest-to-crest arrangement similar to the spring 80 of Fig. 8. However, the wave spring 90 further includes end coils 96 which are bent closed and flat to provide a generally flat contact surface perpendicular to the axis X.

Although the spring 24 of the pedal simulator 14 is schematically illustrated as a single spring 24, the spring 24 can be any suitable spring arrangement or plurality of springs, such as the embodiments of Figs. 3 through 9, which combine to provide a non-linear progressive spring rate characteristic of the spring rate shown in the plot of Fig. 2. For example, a spring assembly can be provided by stacking a plurality of springs 80 of Fig. 8 and placing them between the piston 22 and the end 25 of the housing 18. Note that the total spring rate characteristic for this spring assembly is formed from the series stacked spring rate characteristic, due to the crest-to-crest arrangement between the coils of the individual springs 80, and the parallel stacked spring rate characteristic, due to the adjacent placement of the ends of the springs 80.

There is illustrated in Fig. 10 a simplified hydraulic schematic of a second embodiment of a brake system, indicated generally at 100. The brake system 100 includes a master cylinder 102 for pressurizing brake fluid when the operator of the brake system 100 depresses a brake pedal 104 coupled to the master cylinder 102. A fluid reservoir 106 is in fluid communication with the master cylinder 102 in a conventional fashion and holds a supply of brake fluid

generally at atmospheric pressure. When the master cylinder 102 is actuated by the depression of the brake pedal 104, the pressurized brake fluid produced by the master cylinder 102 enters the brake system 100 from the master cylinder 102 via a conduit 108. The master cylinder 102 is selectively in fluid communication with a wheel brake 110 via the conduit 108.

Although only one wheel brake 110 is shown, it should be understood that the brake system 10 can configured to actuate any number of wheel brakes 110. For example, the master cylinder 102 can be a tandem master cylinder pressurizing a pair of fluid conduits, wherein each fluid conduit is in fluid communication with a pair of wheel brakes associated with a pair of front wheels and a pair of rear wheels, respectively, of a four wheeled vehicle.

A base brake valve 112 is disposed in the conduit 108 and controls the flow of fluid flowing between the master cylinder 102 and the wheel brake 110. The base brake valve 112 can be any suitable valve which can control the flow of fluid in the conduit 108. For example, the base brake valve 112 can be a normally open, solenoid actuated, binary valve movable between an open position 112a and a closed position 112b.

The brake system 100 further includes a source of pressurized fluid or a fluid pressure generator circuit, indicated generally by phantom lines 114. The fluid pressure generator circuit 114 provides pressurized fluid to the brake system 100 via fluid conduit 116 during normal braking to achieve a desired amount of braking, as will be explained in detail below. The fluid pressure generator circuit 114 also provides pressurized fluid for brake operation during ABS (anti-lock braking), TC (traction control), and VSC (vehicle stability control) modes, as will be explained below.

The fluid pressure generator circuit 114 and various solenoid valves of the brake system 100 (further described below) are controlled by an electronic control unit (ECU) 118 using information from various sensors (not all shown)

which monitor various vehicle parameters, such as wheel speed. The brake system 100 includes a pressure transducer 120 in fluid communication with the conduit 108 for transmitting information about the pressure produced by the master cylinder 102 to the ECU 118. Alternatively, the brake system 100 may include a brake pedal displacement transducer 122 operatively connected to the brake pedal 104 to provide a signal indicative of the position of the brake pedal 104. The pressure transducer 120 and/or the brake pedal displacement transducer 122 sense the input force or displacement, respectively, applied by the driver of the vehicle to effectively actuate the fluid pressure generator circuit 114 as demanded by the driver.

The positioning of the base brake valve 112 is controlled by the ECU 118. The base brake valve 112 provides for a "manual push through" to the wheel brake 110, i.e., pressurized fluid from the master cylinder 102 can be sent to the wheel brake 110 when the base brake valve 112 is deenergized, and thus in its open position 112a. However, under normal boosted braking conditions the base brake valve 112 is actuated to its closed position 112b and the fluid pressure generator circuit 114 delivers pressurized fluid through the conduit 116 to actuate the wheel brake 110, as will be discussed in detail below.

The brake system 100 further includes a pedal simulator, indicated generally at 124 to provide compliance for the brake pedal 104 when the base brake valve 112 is closed, as in normal braking operation. As will be explained further below, during normal braking operation as pressurized fluid for braking is supplied to the wheel brake 110 from the fluid pressure generator circuit, the base brake valve 112 is shut to direct the fluid to the wheel brake 110. Thus, fluid pressurized in the master cylinder 102 by operation of the brake pedal 104 must be directed elsewhere in order to provide compliance (i.e., movement) of the brake pedal 104 when acted on by the driver. In the brake system 100, pressurized brake fluid from the master cylinder 102 is directed through the

conduit 108 to the pedal simulator 124. The pedal simulator 124 serves as an accumulator providing a compliance volume for the brake system 100 during normal braking, as will be explained in detail below.

The pedal simulator 124 includes a housing 126 having a bore 128 formed therein. A piston 130 is sealingly slidably disposed in the bore 128. A fluid chamber 132 is defined by the piston 130 and the bore 128. The fluid chamber 132 is selectively in fluid communication with the master cylinder 102 via the conduit 108 via a conduit 134. The pedal simulator 124 further includes a spring 136 interposed between the piston 130 and an end 138 of the bore 128 of the housing 126. The spring 136 can be any suitable spring arrangement having a non-linear spring rate, and more preferably having a non-linear progressive spring rate, in accordance with the present invention. For example, any one or combination of the springs illustrated in Figs. 3 through 9 can be used for the spring 136.

The brake system 100 further includes a pedal simulator valve 140 disposed in the conduit 134 and controls the flow of fluid flowing between the master cylinder 102 and the wheel brake pedal simulator 124. The pedal simulator valve 140 can be any suitable valve which can control the flow of fluid in the conduit 134. For example, the pedal simulator valve 140 can be a normally closed, solenoid actuated, binary valve movable between an open position 140b and a one-way position 134a, wherein fluid is permitted to flow from the pedal simulator 114 to the conduit 134 but not flow in the opposite direction. The positioning of the pedal simulator valve 140 is controlled by the ECU 118. The pedal simulator valve 140 provides for isolation of the pedal simulator 124 by moving to its one-way position 140a, for example, by a loss of electrical power, so that pressurized fluid from the master cylinder 102 can be sent to the wheel brake 110 since the base brake valve 112 is deenergized in its open position 112a. However, under normal boosted braking conditions, the

pedal simulator valve 140 is energized to its open position 140b and the master cylinder 102 delivers pressurized fluid through the conduit 108 to the pedal simulator 140.

The fluid pressure generator circuit 114 includes a pump 144 which is driven by a motor 146. The pump 144 has an inlet 144a which draws fluid from a reservoir 149. Alternatively, the inlet 148a can draw fluid from the reservoir 106 instead of drawing from the reservoir 149. The pump 144 has an outlet 144b which is in fluid communication with the conduit 116. Preferably, the fluid pressure generator circuit 114 includes an accumulator 150 in fluid communication with the conduit 116. The accumulator 150 can be any suitable accumulator structure. For example, the accumulator 150 can include a diaphragm 152 or metal bellows which is biased by a gas, such as nitrogen, to pressurize the fluid stored in a fluid chamber 154. The fluid chamber 154 is in fluid communication with the conduit 116 via a conduit 156. Alternatively, the accumulator 150 can include a spring biased piston for pressurizing fluid stored in the fluid chamber 154. An accumulator valve 158 is disposed in the conduit 156 and controls the flow of fluid between the accumulator 150 and the pump 144. Preferably, the accumulator valve 158 has a first position 158a which is open, and a second position having a one-way check valve 160 which may allow fluid to flow in a direction from the pump 144 to the chamber 154, but restricts the flow of fluid in the opposite direction.

The brake system 100 further includes a pressure control valve 162 which controls the flow of fluid between the fluid pressure generator circuit 114 and the wheel brake 110. As will be explained below, the pressure control valve 162 regulates the generally high pressure from the fluid pressure generator circuit 114 to a desired pressure level as requested by the driver. The pressure control valve 162 is operated by the ECU 118 and can be any suitable valve arrangement, such as for example, a proportional solenoid actuated valve.

The operation of the brake system 100 will now be described. During normal operation of the brake system 100, the pedal simulator valve 140 is actuated to its open position 140b, and the base brake valve 112 is actuated to its closed position 112b. The driver of the vehicle depresses the brake pedal 104, thereby pressurizing the fluid in the master cylinder 102 which is directed to the conduit 108. The pressurized fluid in the conduit 108 is diverted into the pedal simulator 124 via the conduit 134. The fluid enters the chamber 132 expanding the chamber 132, thereby depressing the spring 136 to provide the force feedback. As stated before, the spring 136 can be any suitable spring arrangement having a non-linear spring rate, and more preferably having a non-linear progressive spring rate, in accordance with the present invention. Via information from the pressure transducer 120 and/or brake pedal displacement transducer 122, the ECU controls the fluid pressure generator circuit 114 and the pressure control valve 152 to provide the desired fluid pressure to the wheel brake 110 via the conduit 116 as demanded by the driver. More specifically, the motor 146 and pump 144 are actuated as required to provide generally high fluid pressure to the accumulator 150 via the conduit 156. The accumulator valve 150 is actuated to its open position to supply the high pressurized fluid to the conduit 116. The pressure control valve 162 is then actuated by the ECU 118 to provide the desired fluid pressure to the wheel brake 110.

The brake system 100 also provides manual push-through to the wheel brake in case of faulty operation of the brake system 100. For example, if the brake system 100 is not provided with electrical power, the pedal simulator valve 140 and the base brake valve 112 will be positioned at their normally closed and open positions 140a and 112a, respectively. Fluid is then diverted directly to the wheel brake 110 from the master cylinder 102 via the conduit 108.

There is illustrated in Fig. 11 a simplified hydraulic schematic of a third embodiment of a brake system, indicated generally at 200. The brake system

200 includes a master cylinder 202 operatively connected to a brake pedal 204. A reservoir 206 is in fluid communication with the master cylinder 202 and holds a supply of brake fluid generally at atmospheric pressure. A switch 208 may be connected to the reservoir 206 for sensing the fluid level within the reservoir 206. When the master cylinder 202 us actuated by the depression of the brake pedal 204, the pressurized fluid generated by the master cylinder 202 enters the brake system 200 via a pair of conduits 210 and 212.

The brake system 200 further includes a pressure generator circuit, indicated by the phantom line 214. The fluid pressure generator circuit 214 can be similar to the fluid pressure generator circuit 114 of the brake system 100 of Fig. 10. The fluid pressure generator circuit 214 generally provides pressurized fluid to the brake system 200 via a supply conduit 216. The fluid pressure generator circuit 114 also selectively provides pressurized fluid to the brake system 200 via a high pressure conduit 218, as will be explained below. As will be explained below, the fluid pressure generator circuit 214 provides pressurized fluid to the brake system during normal boosted braking and for brake operations such as anti-lock braking, traction control, and vehicle stability modes.

The fluid pressure generator circuit 214 and various solenoid valves of the brake system 200 are controlled by an electronic control unit (ECU) 220 using information from various sensors, not all shown, which measure various vehicle parameters, such as wheel speed. Preferably, the brake system 200 includes a pressure transducer 222 in fluid communication with the conduit 212 for transmitting pressure information to the ECU 220. Alternatively, the pressure transducer 212 may be located in the conduit 210. The brake system 200 may also include a brake switch 224 connected to the brake pedal 204 to provide a signal that the driver of the vehicle is depressing the brake pedal 204.

The brake system 200 includes a first wheel brake 226 which is in selective fluid communication with the master cylinder 202 via the conduit 212.

A first base brake valve 228 is in fluid communication between the master cylinder 202 and the first wheel brake 226. The first base brake valve 228 is movable between a normally open position 228a and a closed position 228b. Preferably, the first base brake valve 228 is a normally open, pilot operated 2-position, 2-way valve. The positioning of the first base brake valve 228 is regulated by the pressure differential between the a conduit 229 (located between the first isolation valve 264 and the override valves 360 and 362) and the combination of the conduit 212 and a return conduit 230 in fluid communication with the reservoir 206. The first base brake valve 228 provides for a "manual push through" to the first wheel brake 226, i.e., pressurized fluid from the master cylinder 202 can be sent to the first wheel brake 226 when the first base brake valve 288 is in its open position 228a. However, generally under normal boosted braking conditions, the first base brake valve 228 is in its closed position 228b and the fluid pressure generator circuit 214 delivers pressurized fluid through the supply conduit 216 to actuate the first wheel brake 226, as will be discussed in detail below.

The brake system 200 further includes a second wheel brake 232 which is selectively in fluid communication with the master cylinder 202 via the conduit 212. A second base brake valve 234 is in fluid communication between the second wheel brake 232 and the master cylinder 202. The second base brake valve 234 is movable between a normally open position 234a and a closed position 234b. Preferably, the second base brake valve 234 is a normally open, pilot operated 2-position, 2-way valve, the operation of which is regulated by the pressure differential between a conduit 235 (located between the second isolation valve 266 and the override valves 364 and 366) and a combination of the conduit 212 and the return conduit 230. The second base brake valve 234 provides for a "manual push through" to the wheel brake 232, i.e., pressurized fluid from the master cylinder 202 can be sent to the second wheel brake 232

when the second base brake valve 234 is in its open position 234a. However, generally under normal boosted braking conditions, the second base brake valve 234 is in its closed position 234b and the fluid pressure generator circuit 214 delivers pressurized fluid through the supply conduit 216 to actuate the second wheel brake 226, as will be discussed in detail below.

The brake system 200 further includes a fluid conduit 236 which branches off from the conduit 212 adjacent the first wheel brake 226 and is in fluid communication with the first wheel brake 226 and a first fluid separator assembly 238. The first fluid separator assembly 238 includes a spring-biased piston 240 separating first and second chambers 242 and 244, respectively. The second chamber 244 is in fluid communication with the first wheel brake 226 via the conduit 236. The first fluid separator 238 isolates the fluid in the master cylinder 202 and the first wheel brake 226 from the fluid in the fluid pressure generator circuit 214. The fluid pressure generator circuit 214 provides pressurized fluid to the first chamber 242 via a conduit 246 which is selectively in fluid communication with the supply conduit 216. The first fluid separator 238 includes a spring 248 which urges the piston 240 in a direction to minimize the size of the chamber 242.

A conduit 250 branches off from the conduit 212 adjacent the second wheel brake 232 and provides fluid communication between the second wheel brake 232 and a second fluid separator assembly 252. The second fluid separator assembly 252 is similar in structure and function as the first fluid separator assembly 238. The second fluid separator assembly 252 includes a spring-biased piston 254 separating first and second chambers 256 and 258, respectively. The second chamber 258 is in fluid communication with the second wheel brake 232 via the conduit 250. The second fluid separator 252 isolates the fluid in the master cylinder 202 and the second wheel brake 232 from the fluid in the fluid pressure generator circuit 214. The fluid pressure

generator circuit 214 provides pressurized fluid to the first chamber 256 of the second fluid separator assembly via a conduit 260 which branches from the supply conduit 216. The second fluid separator 252 includes a spring 262 which urges the piston 254 in a direction to minimize the size of the chamber 256.

A first isolation valve 264 is located in the conduit 246 between the fluid pressure generator circuit 214 and the first chamber 242 of the first fluid separator assembly 238. A second isolation valve 266 is located in the conduit 260 between the fluid pressure generator circuit 214 and the first chamber 256 of the second fluid separator assembly 252. Preferably, the first and second isolation valves 264 and 266 are normally open 2-position, 2-way solenoid operated valves having first, normally open positions 264a and 266a, respectively, and second, one-way positions 264b and 266b, respectively. The one-way positions 264b and 266b restrict fluid from flowing from the fluid pressure generator circuit 214 via the supply conduit 216 to the first and second fluid separator assemblies 238 and 252, respectively, but may permit fluid to flow in the opposite direction.

A first dump valve 268 is located in a conduit 270 which is in fluid communication with the return conduit 230 and the conduit 246. The first dump valve 268 is positioned to control flow to the reservoir 206 from the first chamber 242 of the first fluid separator assembly 238. A second dump valve 272 is located in a conduit 274 which is in fluid communication with the return conduit 230 and the conduit 260. The second dump valve 272 is positioned to control flow between the reservoir 206 and the first chamber 256 of the second fluid separator assembly 252. Preferably, the first and second dump valves 268 and 272 are normally closed 2-position, 2-way solenoid operated valves having first, closed positions 268a and 272a, respectively, and second open positions 268b and 272b, respectively.

A pair of conduits 276 and 278 provide fluid communication between the supply conduit 216 and a third wheel brake 280 and a fourth wheel brake 282, respectively. A third isolation valve 284 is located in the conduit 276 to control fluid flow between the fluid pressure generator circuit 214 to the third wheel brake 280. A fourth isolation valve 286 is located in the conduit 278 to control fluid flow between the fluid pressure generator circuit 214 and the fourth wheel brake 282. Preferably, the third and fourth isolation valves 284 and 286 are normally open 2-position, 2-way solenoid operated valves having first, normally open positions 284a and 286a, respectively, and second, one-way positions 284b and 286b, respectively. The one-way positions 284b and 286b restrict fluid from flowing from the fluid pressure generator circuit 214 via the supply conduit 216 to the third and fourth wheel brakes 280 and 282, but allow fluid to flow in the opposite direction.

A third dump valve 288 is located in a conduit 290 which is in fluid communication with the return conduit 230 and the conduit 276. The third dump valve 288 controls the flow of fluid between the reservoir 206 and the third wheel brake 280. A fourth dump valve 292 is located in a conduit 294 which is in fluid communication with the return conduit 230 and the conduit 278. The fourth dump valve 292 controls the flow of fluid between the reservoir 206 and the fourth wheel brake 282. Preferably, the third and fourth dump valves 288 and 292 are normally closed 2-position, 2-way solenoid operated valves having first, closed positions 288a and 292a, respectively, and second, open positions 288b and 292b, respectively.

Preferably, the first and second wheel brakes 226 and 232 are associated with the front wheels of the vehicle in which the brake system 200 is installed, and the third and fourth wheel brakes 226 and 232 are associated with the rear wheels. However, the wheel brakes of the brake system 200 can be connected in any suitable arrangement.

Although the brake system 200 is shown having independent circuits for the wheel brakes 280 and 282, the brake system 200 could be adapted to include a single isolation valve and a single dump valve for the wheel brakes 280 and 282, such as for example, if the wheel brakes 280 and 282 are rear wheel brakes on a front wheel driven vehicle. Alternatively, the brake system 200 could be adapted to include isolation valves, dump valves, and fluid separator assemblies for each of the wheel brakes 280 and 282. The brake system 200 could also be adapted to include a single isolation valve, a single dump valve, and a single fluid separator assembly to supply both of the wheel brakes 280 and 282.

The fluid pressure generator circuit 214 includes a pump 300 which is driven by a motor 302. The pump 300 has an inlet 300a in fluid communication with the reservoir 206 via a conduit 301, and an outlet 300b in fluid communication with the high pressure conduit 218. The pump 300 operates to draw fluid from the reservoir 206 and supply the fluid at an increase pressure to the conduit 218. A check valve 304 may be provided at the outlet 300b to help prevent the flow of fluid from the conduit 218 into the outlet 300b of the pump 300. Preferably, the fluid pressure generator circuit 214 includes a pressure transducer 306 in fluid communication with the conduit 218 for sensing the pressure therein. Alternatively, a pressure switch can be used in place of the pressure transducer 306. An accumulator 308 is in fluid communication with the pump outlet 300b through the check valve 304. The accumulator 308 can be any suitable accumulator structure. For example, the accumulator 308 can include a diaphragm or metal bellows which is biased by a gas, such as nitrogen, to pressurize the fluid stored in the accumulator 308. Alternatively, the accumulator 308 can include a spring biased piston for pressurizing fluid stored in the accumulator 308.

The fluid pressure generator circuit 214 further includes a boost valve, indicated generally at 310. As will be described in detail below, the boost valve

310 is preferably mechanically actuated by a pedal travel simulator, indicated generally at 312. Preferably, the boost valve 310 is a 3-position, 3-way valve. The boost valve 310 can have any suitable valve arrangement, such as a poppet or spool valve. The boost valve 310 is generally located between the pump output 300b and the supply conduit 216. The boost valve 310 is also in fluid communication with the master cylinder 202, but it is mechanically decoupled from it. That is, the boost valve 310 is not mechanically connected to the master cylinder 202 or the brake pedal 204. Therefore, the boost valve 310 can be located remotely from the master cylinder 202, providing flexibility in positioning the boost valve 310 within the vehicle. The boost valve 310 is connected at a first port 310a to the outlet of the pump 300b via the conduit 218. A second port 310b of the boost valve 310 is connected to the reservoir 206 via a conduit 314, a conduit 328, and the conduit return 230. A third port 310c of the boost valve 310 is selectively in fluid communication with the wheel brakes 226, 232, 280, and 282 via the supply conduit 216.

The boost valve 310 generally has a first position 316a, a second position 316b, and a third position 316c. In the first position 316a, the second port 310b is in fluid communication with the third port 310c. In the second position 316b, all three ports 310a, 310b, and 310c are disconnected from each other. In the third position 316c, the first port 310a is connected to the third port 310c. The operation of the boost valve 310 will be explained in detail below.

The boost valve 310 preferably has an integral pressure relief function, represented schematically at 318 which selectively permits fluid communication between the outlet 300b of the pump 300 and the reservoir 206, via the conduit 218 and the return conduit 230. The pressure relief function 318 limits the output pressure of the pump 300 by opening at a predetermined pressure differential to create a return path between the pump outlet 300b and the

reservoir 206. The pressure relief valve can also be a separate component of the brake system 200.

The pedal travel simulator 312 is an accumulator which receives brake fluid from the master cylinder 202, as will be explained in detail below. The pedal travel simulator 312 can be designed to provide the driver with a pedal feel that is similar to typical hydraulic braking systems using conventional boosters or any other desirable pedal feel. The pedal travel simulator 312 includes a bore 320 having a piston 322 slidably disposed therein. The piston 322 divides the bore 320 into first and second chambers 324 and 326. The first chamber 324 is selectively in fluid communication with the master cylinder 202 via the conduit 210. The second chamber 326 is in fluid communication with the reservoir 206 via a conduit 328. The pedal travel simulator 312 includes a pair of springs 340 and 342. The first spring 340 is interposed between the piston 322 and the boost valve 310. As will be explained below, the first spring 340 actuates the boost valve 310. The second spring 342 is interposed between the piston 322 and a fixed structure of the body of the pedal travel simulator 312.

Preferably, one or both of the springs 340 and 342 have a progressive non-linear spring rate similar to the spring 24 of the brake system 10 of Fig. 1, such as the embodiments of the springs described and shown in Figs. 3 through 9. The operation of the boost valve 310 and the pedal travel simulator 312 will be described below.

Located in the conduit 210 between the master cylinder 202 and the first chamber 324 of the pedal travel simulator 312 is a pedal travel simulator valve 350. Preferably, the pedal travel simulator valve 350 is a normally closed 2-position, 2-way pilot operated valve, however, any suitable valve can be used, such as a solenoid actuated valve. The pedal travel simulator valve 350 preferably has a first position 350a having a one-way check valve 352 which may allow fluid to flow in a direction from the first chamber 324 of the pedal

travel simulator 312 into the conduit 210, but restricts the flow of fluid in the opposite direction. The pedal travel simulator valve 350 also has a second position 350b which is open. The pedal travel simulator valve 350 is biased to the first position 350a by a spring 354. The pilot operated pedal travel simulator valve 350 senses the pressure differential between the conduits 210 and 218. The pedal travel simulator valve 350 moves to the second open position 350b when the fluid pressure in the conduit 218 from the high pressure accumulator 308 overcomes the pressure in the first chamber 324 and the force of the spring 354 biasing the pedal travel simulator valve 350 to the first position 350a. Thus, when the pedal travel simulator valve 350 is in the first position 350a, fluid flowing from the master cylinder 202 is blocked so that the pressurized fluid from the master cylinder 202 will flow into the wheel brakes 226 and 232 through the first and second base brake valves 228 and 234, respectively.

A bleed screw port 358 is preferably provided, such as an integral structure of the pedal travel simulator valve 350, for bleeding and filing various components of the brake system 200. The bleed screw port 358 provides for a fluid path from the high pressure conduit 218, through the bleed screw port 358, through the pedal travel simulator valve 350 and into the master cylinder 202.

The pedal travel simulator valve 350 is provided for certain conditions in which there is generally not enough fluid pressure from the high pressure accumulator 308 for proper operation of the boost valve 310. In this condition, the pedal travel simulator valve 350 will close off fluid communication between the master cylinder 202 and the pedal travel simulator 312, and the first and second base brake valves 228 and 234 will provide for manual push through to the first and second wheel brakes 226 and 232.

The brake system 200 includes a first override dump valve 360 and a first override isolation valve 362 for regulating the flow of pressurized fluid from the conduit 218 of the fluid pressure generator circuit 214 to the wheel brakes 226

and 280, such as for example, during a traction control (TC) or vehicle stability control (VSC) brake operation. The first override dump valve 360 is movable between a closed position 360a and an open position 360b. The first override isolation valve 362 is moveable between an open position 362a and a one-way position 362b, wherein the fluid is permitted to flow from the supply conduit 216 to the conduits 246 and 276. The brake system 200 further includes a second override dump valve 364 and a second override isolation valve 366 for regulating the flow of pressurized fluid from the conduit 218 of the fluid pressure generator circuit 214 to the wheel brakes 232 and 282. The second override dump valve 364 is movable between a closed position 364a and an open position 364b. The second override isolation valve 366 is moveable between an open position 366a and a one-way position 366b, wherein the fluid is permitted to flow from the supply conduit 216 to the conduits 260 and 278.

By providing two pairs of override dump and isolation valves, one of the pairs of wheel brakes can be subjected to the high pressure in the conduit 218 by actuating the associated override dump and isolation valves, while the other pair of wheel brakes can simultaneously be controlled by the boost valve 310. The brake system 200 is preferably configured so that the wheel brakes 226 and 280 are grouped together, while the wheel brakes 232 and 282 are grouped together.

Preferably, the brake system 200 further includes a compliance accumulator 370 having a spring biased piston 372 pressurizing a chamber 374. The chamber 374 is in fluid communication with the third port 310c of the boost valve 310. The compliance accumulator 370 generally supplies initial instantaneous flow to the supply conduit 216 until the boost valve 310 can respond with sufficient flow. For example, if the boost valve 310 is designed with an internal dampening system for a valve, such as a poppet valve, to help stabilize the valve, the compliance accumulator can provide sufficient flow at a relatively quick response rate when the isolation and dump valves are pulsed to

generally provide instantaneous pressure for the brake system 200. If desired, the compliance accumulator 370 can be omitted from the brake system 200.

The operation of the brake system 200 shall now be described. During normal boosted braking operation, the driver of the vehicle in which the brake system 200 is installed, will depress the brake pedal 204 to actuate the wheel brake 226, 232, 280 and 282. The term "normal boosted braking" refers to the operation of the brake system 200, wherein the ignition system of the vehicle is on and the brake system 200 has not entered into an ABS, TC, or VSC operation. Movement of the brake pedal 204 moves pistons (not shown) within the master cylinder 202, thereby pressurizing the fluid within the conduits 210 and 212. During operation of the vehicle and based on information from the pressure transducer 306, the pump 300 is actuated to supply relatively high fluid pressure to the accumulator 308 and the conduit 218 within a selected pressure range. Generally, the fluid pressure within the conduit 218 will be greater than the fluid pressure within the conduits 210 and 212 even when the master cylinder 202 is actuated. The differential pressure across the pedal travel simulator valve 350 causes the pedal travel simulator valve 350 to move to its open position 350b. Thus, the pressurized fluid within the conduit 210 from the master cylinder 202 flows through the conduit 210 and into the pedal travel simulator 312. The pressurized fluid from the conduit 210 expands the first chamber 324, thereby advancing the piston 322 leftward, as viewing Fig. 11. The movement of the piston 322 acts to compress the spring 340 against a portion of the boost valve 310, thereby actuating the boost valve 310 to the third position 316c. The spring 340 and 342 of the pedal travel simulator 312 can be designed to provide the driver with a pedal feel, or a reactionary force acting on the brake pedal 204, which is similar to typical brake systems, or the pedal travel simulator 312 can be designed so as to create any suitable desirable reactionary force.

If desired, the pedal travel simulator 312 can be configured so that the spring 342 is not engaged until the piston 322 has traveled a predetermined distance.

In the third position 316c, the boost valve 310 generally allows pressurized fluid from the pump 300 and the high pressure accumulator 308 to flow from the conduit 218 into the supply conduit 216. The fluid pressure generator circuit 214 operates to supply fluid pressure to the supply conduit 216, referred herein as "boost pressure", at a "variable boost ratio" in relation to the pressure generated from the master cylinder 202, sensed via the spring 340 of the pedal travel simulator 312. The boost ratio is variable and changes in relation to the displacement of the brake pedal 204 because of the progressive non-linear spring rate of the springs 340 and/or 342. The boost pressure is generally greater than the pressure within the conduits 210 and 212 by a variable gain, which multiple is termed the "variable boost ratio". For example, if the pressure within the conduits 210 and 220 are at 100 p.s.i.g. and the boost ratio is 6, the boost valve 310 will supply fluid at about 600 p.s.i.g. to the supply conduit 218. The boost valve 310 will shuttle to its second position 316b to close the fluid communication between the conduit 218 and the supply conduit 216 when the pressure of the fluid in the supply conduit 216 is approximately equal to the pressure supplied by the master cylinder 202 multiplied by the boost ratio.

Since the pressure within the conduits 210 and 212 will generally be lower than the pressure within the supply conduit 216, the pilot operated first and second base brake valves 228 and 234 are shuttled to their respective closed positions 228b and 334b, thereby closing direct fluid communication between the conduit 212 and the wheel brakes 226 and 234, respectively. With the first and second base brake valves 228 and 234 closed, the fluid pressures at the wheel brakes 236 and 232 can exceed the fluid pressure generated by the master cylinder 202 within the conduit 212.

During normal boosted braking, the isolation valves 284 and 286 are in their open positions 284a and 286a to permit the flow of fluid from the supply conduit 216 to the wheel brakes 280 and 282. The dump valves 288 and 292 are also typically in their closed position 288a and 292a during normal boosted braking to prevent fluid from entering the return conduit 230 to the reservoir 206. The pressurized fluid in the supply conduit 216 also flows through the open isolation valves 264 and 266 during normal boosted braking and into the first chambers 242 and 256 of the first and second fluid separator assemblies 238 and 252, respectively. The pressurized fluid moves the pistons 240 and 254 in the first and second fluid separator assemblies 238 and 252, respectively, towards the second chambers 244 and 258, thereby pressurizing the fluid therein. The pressurized fluid flows from the second chambers 244 and 258 into the wheel brakes 226 and 232 via the conduits 236 and 250 to brake the vehicle. Note that during normal boosted braking, the dump valves 268 and 272 are typically in their respective closed positions 268a and 272a to prevent fluid from entering the return conduit 230 to the reservoir 206.

Based on information from the pressure transducer 306, the ECU 220 may actuate the motor 302 of the pump 300 during normal boosted braking to supply relatively high pressure to the conduit 218 and the high pressure accumulator 308 within a selected pressure range. The boost valve 310 will shuttle between its positions 316a, 316b, and 316c to maintain the pressure in the supply conduit 216 at a pressure which is generally equal to the pressure within the conduits 210 and 212 generated by the master cylinder 202 multiplied by the boost ratio.

When the driver releases the brake pedal 204, the fluid within the first chamber 324 of the pedal travel simulator 312 returns to the master cylinder 202 through the pedal simulator valve 350 in its open position 350b. The boost valve 310 is shuttled to the first position 316a which allows fluid communication

between the supply conduit 216 and the conduit 328 to the reservoir 206. The pressurized fluid in the first chambers 242 and 256 of the first and second fluid separator assemblies 238 and 252, respectively, returns to the reservoir 206 by flowing through the conduits 246 and 260, respectively, and then through the supply conduit 216, the boost valve 310, and the conduit 328. As the fluid separator assembly pistons 240 and 254 move towards the first chambers 242 and 256, the fluid pressure in the wheel brakes 226 and 232, respectively, is reduced and the fluid therein returns to the second chambers 244 and 258, respectively. The first and second base brake valves 228 and 234 are shuttled to the open positions 228a and 234a, respectively, as the pressure in the supply conduit 216 generally drops below the ratio pressure supplied by the master cylinder 202. It should be understood that this is a generalization and that the pressure generated by the master cylinder 202 can be higher or lower than the pressure in the supply conduit 216 from the boost valve 310. Any residual pressurized fluid in the wheel brakes 226 and 232 flows back to the master cylinder 202 via conduit 212. The pressure within the wheel brakes 280 and 282 is reduced, and the fluid therein returns to the reservoir 206 via the conduits 276 and 278, respectively, and then through the supply conduit 216, the boost valve 310, and the conduit 328.

During an ABS event, the brake system 200 admits pressurized fluid into the supply conduit 216 in a similar manner as during normal boosted braking. However, during an ABS braking operation, the ECU 220 controls the isolation valves 264, 266, 284, and 286 and the dump valves 268, 272, 288, and 292 to regulate the pressure to the wheel brakes 226, 232, 280, and 282, respectively. For example, if the ECU 220 detects that the wheel corresponding to the wheel brake 226 begins to slip appreciably during braking, an ABS dump mode may be entered into. The pressure at the wheel brake 226 is reduced to allow the wheel to spin back up to near vehicle speed. To reduce the pressure at the wheel brake

226, the isolation valve 264 is shuttled to the second, one-way position 264b, such as by actuating the solenoid of the isolation valve 264. When the isolation valve 264 is in the one-way position 264b, fluid in the supply conduit 216 is prevented from reaching the fluid separator assembly 238. The dump valve 268 is shuttled to the open position 268b by actuating the solenoid thereof, thereby allowing the pressurized fluid in the first chamber 242 of the first fluid separator assembly 238 to flow back to the reservoir 206 via the return conduit 230. The brake system 200 may enter into an ABS hold mode to give the wheel time to spin back up to speed. During the ABS hold mode, the pressure at the wheel brake 226 is generally held constant by keeping the isolation valve 264 shuttled to its one-way position 264b and keeping the dump valve 268 in its closed position 268a.

When the ECU 220 detects that the wheel associated with the wheel brake 226 spins back up to near vehicle speed, an ABS apply mode may be entered into in which pressure is increased at the wheel brake 226. The isolation valve 264 is shuttled to the open position 264a and the dump valve 268 is shuttled (or remaining in) its closed position 268a. This allows the pressurized fluid in the supply conduit 216 to expand the first chamber 242 of the fluid separator assembly 238. The expansion of the first chamber 242 causes the piston 240 to move to pressurize the fluid in the second chamber 244, thereby supplying pressurized fluid to the wheel brake 226. The brake system 200 may enter the dump, hold, and apply modes several times during a single ABS event.

When a driven wheel begins to slip during acceleration, the brake system 200 may enter into a traction control (TC) mode. The slipping wheel is braked to slow the wheel and regain traction for improved vehicle acceleration and stability. During a TC mode, the ECU 220 actuates the fluid pressure generator circuit 214 to provide pressurized fluid to the supply conduit 216 and controls the operation of the first and second override isolation valves 362 and 366 and

the first and second override dump valves 360 and 364. The ECU 220 actuates the appropriate override isolation valves 362 and 366 to their one-way positions 362b and 366b, thereby preventing flow from the supply conduit 216 (adjacent the boost valve 310). The ECU 230 also actuates the appropriate override dump valves 360 and 364 to their open positions 360b and 364b, thereby allowing high pressure fluid from the conduit 218 to enter the wheel brakes supply conduit 216. Various ones of the isolation valves 264, 266, 284, and/or 286 and the dump valves 268, 272, 288, and/or 292 may be controlled to brake the slipping wheel to regain traction. More preferably, the override valve 360, 362, 364, and 366 are pulsed and the appropriate isolation valves are actuated. In this scenario, the dump valves may not be actuated at all. For example, if the wheel corresponding to the first wheel brake 226 is a driven wheel and slippage is detected, the first override isolation valve 362 to its one-way position 362b, and actuates the first override dump valve 360 to its open position 360b. The isolation valve 284 corresponding to the other wheel is actuated to its one-way closed position 284b. The isolation valve 264 is optionally pulsed from the one-way position 264b to the open position 264a. More preferably, the override valves are cyclically pulsed. The pressurized fluid in the conduit 246 expands the first chamber 242 of the first fluid separator assembly 238. The expansion of the first chamber 242 moves the piston 240, thereby pressurizing the fluid in the second chamber 244 to provide pressurized fluid to the first wheel brake 226. A traction control hold mode may be entered to keep the pressure constant at the first wheel brake 226. During traction control hold mode, preferably, the override valve 360 is shuttled to its closed position 360a. When the speed of the driven wheel associated with the wheel brake 226 has been reduced to near the vehicle speed, a traction control dump mode may be entered into to reduce the brake pressure at the first wheel brake 226. During a traction control dump mode, the override valve 362 is shuttled to (or remains in) the one-way position

362b. The override valve 362 is then shuttled to its open position 362a. The pressurized fluid in the first chamber 242 can flow out through the dump valve 268 and back to the reservoir 206 via the return conduit 230. After the traction control event has ended, the override dump valve 360 is actuated to stop pulsing and the override isolation valve 362 is moved to its open position 362a allowing the fluid in the supply conduit 216 (located adjacent the boost valve 310) to return to the reservoir 206.

During a VSC event, braking may be required on one or more wheels to improve cornering stability of the vehicle. The driver may or may not be braking at that time, and the braking pressures required may exceed the pressure generated by the master cylinder 202. The brake system 200 is actuated and controls the operation of the override isolation valves 362 and 366 and the override dump valves 360 and 364 in a similar manner as during a TC event, as described above. The isolation and dump valves corresponding to the wheel brakes which are to be actuated control the wheel brake pressure to achieve the desired braking effect or more preferably, the isolation valves are selectively actuated and the override valves are pulsed.

In the embodiment of the brake system 200 as described above, each wheel brake 226, 232, 280, and 282 can be independently modulated from the others above or below the pressure generated by the master cylinder 202, thereby enabling the brake system 200 to be able to perform other braking related functions, such as for example, adaptive cruise control, hill-hold, cruise control with downhill augmentation, panic brake assist, and other related braking functions.

The brake system 200 may also be used to provide Dynamic Rear Proportioning (DRP), or any other multi-wheel combination. For example, the brake system 200 can enter into a DRP mode so that the braking pressures on the front and rear brakes are separately controlled to achieve greater braking

performance at the front and rear axles. For example, the brake system 200 can be configured such that the wheel brakes 226 and 232 are associated with the front wheels, and the wheel brakes 280 and 282 are associated with the rear wheels. The respective isolation valves are regulated to increase or decrease the braking pressure at the wheels of the rear brakes at a different pressure from the front brakes, as required to achieve maximum braking effort with minim wheel slippage.

There is illustrated in Fig. 12 an alternate embodiment of a base brake valve, indicated generally at 400, which can be used for example, as the base brake valves 228 and 234 of the brake system 200 of Fig. 11. The valve 400 includes a housing 402 having a bore 404 formed therein. A main body 406 is disposed in the bore 404. The body 406 is retained in the bore 404 by a retaining ring 408. The body 406 has a stepped axial bore 410 formed therethrough. The body has three radially extending passageways 412, 414, and 416 formed therein. A first end plug 418 is threadably attached to the right-hand end of the body 406, as viewing Fig. 12. The first end plug 418 has an axial bore 420 formed therethrough forming a ball seat 422. A plunger 424 is slidably disposed in the bore 410 of the body 406. The plunger 424 has an axial bore 426 formed therethrough, and a (optional) radially extending passageway 428 formed therethrough. A ball 430 is press-fit into the bore 426 of the plunger 424. The ball 430 and the ball seat 422 Of the first end plug 418 cooperate to form a valve member. A spring 432 biases the plunger 424 and the ball 430 away from the ball seat 422. The plunger 424 includes a seal 434 disposed about the plunger 424 which sealingly engages a portion of the bore 410 of the body 406.

An intermediate body 436 is disposed in the bore 410 of the body 406. The intermediate body 436 includes a seal 438 disposed thereabout which sealingly engages a portion of the bore 410 of the body 406. A second end plug

440 is threadably engaged with the body 406 and seals off one end of the bore 410.

A first chamber 442 is generally defined by the right-hand end of the first end plug 418 and the end of the bore 404 of the housing 402. The first chamber 442 is in fluid communication with a master cylinder, such as the master cylinder 202 of the brake system 200 of Fig. 11 via the conduit 212. A second chamber 444 is generally defined by the first end plug 418, the plunger 424, the ball 430, and the seal 434. The second chamber 444 is in fluid communication with a wheel brake, such as the wheel brake 226 or 232 of the brake system 200, via a conduit 445. A third chamber 446 is generally defined by the plunger 424, the intermediate body 426, and the bore 436 of the plunger 424. The third chamber 446 is in fluid communication with a reservoir, such as the reservoir 206 of the brake system 200 via the conduit 230. A fourth chamber 448 is generally defined by the second end plug 440, the intermediate body 436, and the bore 410 of the body 406. The fourth chamber 448 is in fluid communication with a boost supply conduit, such as the supply conduit 216 (located between the isolation valve 264 or 266 and the override valves) from the boost valve 310 of the brake system 200.

The operation of the base brake valve 400 will now be discussed in cooperation with the brake system 200 of Fig. 11. The valve 400 as shown in Fig. 12 is in the position 228a of the base brake valve 228. At this position, the ball 430 is unseated from the seat 422, thereby allowing fluid communication between the master cylinder 202 and the wheel brake 226. During normal boosted braking, the pressure in the fourth chamber 448 from the conduit 216 is greater than the pressure within the first chamber 442 from the master cylinder 202, thereby biasing the ball 430 against the seat 422 closing of fluid communication between the master cylinder 202 and the wheel brake 226.

Note that the sealing diameter of the seal 438 of the intermediate body 436 is greater than the sealing diameter of the seal 434 of the plunger 424. This difference in sealing diameters causes the plunger to be biased against the seat 422 by a ratio proportional to the ration of sealing areas. It is advantageous to bias the ball 430 against the seat 422 to help reduce the likelihood that the pressure generated by the master cylinder 202, such as by a hard brake apply, will cause the ball 430 to lift from the seat 422 causing the flow of fluid from the conduit 212 to flow directly to the wheel brake 226 via the conduit 445.

Preferably, the sealing diameter of the seal 434 of the plunger 424 is relatively equal to the diameter of the seat 422 so that the pressure at the wheel brake does not sufficiently effect operation of the valve 400. However, it may be desirable for the diameter of the seat 422 to be slightly smaller than the sealing diameter of the seal 434 so that the ball 430 is slightly biased away from the seat 422 by brake pressure. Note that the fluid communication between the third chamber 446 and the reservoir 206 permits relatively unhindered movement of the plunger 424 and the intermediate body 436 within the bore 410 of the body 406 by allowing the third chamber 446 to expand and contract.

There is illustrated in Fig. 13 an alternate embodiment of a pedal travel simulator valve, indicated generally at 500, which can be used for example, as the pedal travel simulator valve 350 of the brake system 200, illustrated in Fig. 11. The valve 500 includes a housing 502 having a stepped bore 504 formed therein. A main body 506 is disposed in the bore 504. The body 506 has a stepped axial bore 508 formed therethrough forming a shoulder defining a ball seat 510. The body 506 also includes a pair of radial passageways 512 and 514 formed therethrough. The passageways 512 and 514 are in fluid communication with the axial bore 508. The body 506 is sealed against the wall of the bore 504 of the housing by a lip seal 516.

The valve 500 further includes a cap 518 fixed to the right-hand end of the body 506, as viewing Fig. 13. The cap 518 has an axial bore 520 formed therethrough. A pin 522 is slidably disposed in the bore 520 of the cap 518 and extends partially into the axial bore 508 of the body 506. The pin 522 is sealingly engaged against the wall of the bore 520 by a seal 524 and back-up ring 526.

An end plug 528 is threadably fastened to the housing 502 and generally closes off the bore 504 of the housing 502. The end plug 528 is fixably attached to the body 506. A ball 530 is disposed in the axial bore 508 of the body 506 and is biased against the ball seat 510 by a spring 532.

A first chamber 534 is generally defined by the right-hand end of the pin 522, as viewing Fig. 13, and the wall of the closed end of the bore 504. The first chamber is in fluid communication with a high pressure source, such as for example, the fluid pressure generator circuit 214 via the conduit 218. A second chamber 536 is generally defined by the wall of the bore 504, the pin 522, the axial bore 508 of the body 506, the radial passageway 514, and the ball seat 510. The second chamber 536 is in fluid communication with a fluid chamber of a pedal travel simulator, such as the chamber 324 of the pedal travel simulator 312 of the brake system 200 of Fig. 11, via a conduit 538. A third chamber 540 is generally defined by the wall of the bore 504 of the housing 502, the axial bore 508 of the body 506. The third chamber 540 is in fluid communication with a master cylinder, such as the master cylinder 202 of the brake system 200 of Fig. 11, via the conduit 210.

The operation of the pedal travel simulator valve 500 will now be described in cooperation with the brake system 200 of Fig. 11. The valve 500 as illustrated in Fig. 13, is in a position corresponding to the position 350a of the pedal travel simulator valve 350. At this position, the ball 530 is seated on the ball seat 510. As fluid enters the first chamber 534 from the conduit 218 from

the fluid pressure generator circuit 214, the pin 522 is advanced leftward, as viewing Fig. 13. Note that during normal boosted braking, the pressure within the first chamber 534 from the fluid pressure generator circuit is generally greater than the pressure within the third chamber 540 from the master cylinder 202 due to the boost ratio. Continued movement of the pin 522 causes the pin 522 to lift the ball 530 from the seat 510, thereby allowing fluid to flow between the second and third chambers 536 and 540 via the passageways 512 and 514. The valve 500 is then in the position corresponding to the position 350b of the pedal travel simulator valve 350.

Note that if the fluid pressure is sufficient, the fluid can flow in a direction from the second chamber 536 to the third chamber 540 around the lip seal 516. This lip seal arrangement corresponds to the check valve 352 schematically illustrated in the pedal travel simulator valve 350 of the brake system 200 of Fig. 11.

The bleed screw port 358 schematically represented in Fig. 11 of the brake system 200 is provided by a valve seat 542 formed in the bore 504 of the housing 502 which cooperates with cap 518. During bleeding of the brake system 200, the end plug 528 is threadably positioned so that a gap exists between the valve seat 542 and the cap 518. After the brake system has been sufficiently bled, the end plug 528 can be advanced until the cap 518 seats against the valve seat 542.

Referring now to Fig. 14, there is shown an alternate embodiment of a boost valve, indicated generally at 550, which can be used for the boost valve 310 of the brake system 200 illustrated in Fig. 11. The boost valve 550 is mechanically actuated by a pedal travel simulator, indicated generally at 552.

The boost valve 550 and the pedal travel simulator 552 are housed in a valve body 554. The valve body 554 has a stepped bore 556 formed therein

defining an axis Y. Generally, the components of the boost valve 550 and the pedal travel simulator 552 are co-axially aligned along the axis Y.

The pedal travel simulator 552 includes an end plug 557 closing off one end of the bore 556 by a threaded connection therebetween. A sleeve 558 is disposed in the bore 556 of the body 554 and is threadably engaged therewith. The sleeve 558 has a through bore 559 formed therethrough. A two-piece piston 560 is slidably disposed within the bore 559 of the sleeve 558. The piston 560 has a first portion 560a and a second portion 560b which are fixably attached together, such as by a threaded connection or press-fit. The second portion 560b extends from the first portion 560a and extends through the bore 559 of the sleeve 558. The first portion 560a of the piston 560 is sealingly engaged with the wall of the bore 559 by a seal 564 disposed about the first portion 560a. The piston 560 is biased leftward, as viewing Fig. 14, by a spring 568. A spring 569 is disposed between the first portion 560a and a shoulder 570 radially extending outwardly from the second portion 560b. Preferably, the spring 569 has a progressive non-linear spring rate. For example, the spring 569 can be a plurality of wave springs, such as the wave springs 80 illustrated in Fig. 8. In the position shown in Fig. 14, and end 569a of the spring 569 is spaced from an inwardly extending shoulder 570 by a distance "D". The left-hand end of the spring 569, as viewing Fig. 14, axially rests against a shoulder 571 formed in the second portion 560b. The seal 564, the piston 560, and the closed end of the bore 556 of the end plug 558 generally define a first fluid chamber 572. The first fluid chamber 572 is in fluid communication with a control input port, such as the conduit 210 from the master cylinder 202 of the brake system 200 of Fig. 11.

The pedal travel simulator 552 further includes a spring 576 which biases the piston 560 rightward, as viewing Fig. 14. Preferably, the spring 576 has a progressive non-linear spring rate characteristic. For example, the spring 576

can be a helical coil compression spring formed from a wire having a constant diameter or cross-sectional shape, but having a variable pitch. Of course, any suitable springs can be used in place of the springs 569 and 576, such as any of the embodiments shown in Figs. 3 through 9.

The right-hand end of the spring 576 engages a cup shaped first spring retainer 578. An annular seat 582 is formed in the first spring retainer 578. A rounded end 584 is formed in the left-hand end of the second portion 506b of the piston 560. The rounded end 584 engages the annular seat 582 of the of the first spring retainers 578. The annular seat 582 and the rounded end 584 cooperate to assist in axially aligning the force transmitted between the piston 560 and the first spring retainer 578. Of course, any suitable arrangement between the piston 560 and the first spring retainer 578 can be provided. As shown in Fig. 14, the left-hand end of the spring 576 engages a cup shaped second spring retainer 588 having an annular seat 592 formed therein.

The boost valve 550 includes a sleeve 594 sealingly disposed within the bore 556 of the housing 554. The sleeve 594 is retained in the bore 556 by a threaded connection therebetween. The sleeve 594 has a bore 598 formed in the right-hand end thereof. Disposed within the bore 598 is a second sleeve 600 having a through bore 602 formed therethrough. The second sleeve 600 can be threadably fastened to the sleeve 594 to adjust the position therebetween. Slidably disposed within the bore 602 of the second sleeve 600 is a reaction spool 604 having a through bore 605. A cap 606 is fastened to the reaction spool 604. The cap 606 has a rounded end 608 formed in the cap 606. The rounded end 608 cooperates with the annular seat 592 of the second spring retainer 588 to assist in axially aligning the force transmitted between the second spring retainer 588 and the cap 606. A spring 610 biases the cap 606 and the reaction spool 604 rightward, as viewing Fig. 14.

A second fluid chamber 612 is generally defined by the bore 556 of the housing 554, the piston 560, the end plug 558, the first and second sleeves 594 and 600, and the reaction spool 604. The second fluid chamber 612 is in fluid communication with a reservoir port, such as the conduit 328 of the brake system 200 of Fig. 11. The second fluid chamber 612 is also in fluid communication with the bore 605 of the reaction spool 604 via an axial bore 613 and radial passageways 615 formed in the cap 606.

As best shown in Fig. 15, the boost valve 550 includes an annular collar 614 fixably attached to the reaction spool 604, such as by a press fit. The annular collar 614 is disposed in a recess 616 formed in the left-hand end of the second sleeve 600, as viewing Figs. 14 and 15. The annular collar 614 includes a longitudinal passageway 618 formed therethrough having flow restrictive orifices 620. The boost valve 550 may include an optional damping chamber 622 generally defined by the recess 616, the reaction spool 604, and the right-hand end of the annular collar 614. The damping chamber 622 functions to assist in dampening the motion of the reaction spool 604 to prevent the formation of undesirable hydraulic pulses which travel through the fluid within the boost valve 550 and also helps eliminate pedal feedback or pulsation to the driver.

As best shown in Fig. 15, the boost valve 550 further includes a valve assembly, generally indicated at 624, which is generally positioned within the first sleeve 594 for limited axial movement. The valve assembly 624 includes first and second cylindrical discs 626 and 628, which are axially spaced apart from each other. The first and second cylindrical discs 626 and 628 are similar in construction, and are preferably identical and reversible. The first and second discs 626 and 628 include axial bores 626a and 628a, respectively, extending therethrough. The first and second discs 626 and 628 each include a pair of openings 626b and 628b, respectively, formed therethrough which are spaced

apart from each other generally across the axial bores 626a and 628a, respectively.

The second disc 628 has a valve member, such as a first ball 630 press fit into the axial bore 628a of the second disc 628 and extending beyond the right-hand end of the second disc 628. The first ball 630 cooperates with a valve seat 632 formed in the reaction spool 604 to regulate the flow of fluid through the bore 605 of the reaction spool 604. Preferably, the first ball 630 is press fit from the left-hand end of the axial bore 628a to its position as shown in Figs. 14 and 15. A second ball 634 may be press fit into the axial bore 628a of the second disc 628 to act as a stop to help prevent the first ball 630 from dislodging through the axial bore 628a. Preferably, the diameter of the second ball 634 is slightly greater than the diameter of the first ball 630.

The first sleeve 594 has a pair of bores 636 formed therethrough extending between the bore 598 and a recess 638 formed in the left-hand end of the first sleeve 594. A central bore 640 is formed in the first sleeve 594 between the pair of bores 636 and forms a valve seat 642. The central bore 640 is in fluid communication with a radial bore 644 formed through the first sleeve 594.

The first disc 626 has a valve member, such as a first ball 646 press fit into the axial bore 626a of the first disc 626 and extending beyond the right-hand end of the first disc 626. The first ball 646 cooperates with the valve seat 642 formed in the first sleeve 594 to regulate the flow of fluid through the central bore 640 and the radial bore 644. A second ball 648 is press fit into the axial bore 626a of the first disc 626 and extends beyond the left-hand end of the first disc 626.

The valve assembly 624 further includes a pair of cylindrical spacers 650 which are positioned between the first and second discs 626 and 628. The spacers 650 extend through the pair of bores 636 formed through the first sleeve 594, thereby allowing limited axial movement of the valve assembly 624 with

respect to the first sleeve 594. The ends of the spacers 650 can be fixably attached to the first and second discs 626 and 628 by pressing a ball 652 into a bore 654 formed in the end of the spacers 650 to expand the tubular shaped wall of the end of the spacers 650 radially outwardly, thereby achieving a press fit between the end of the spacers 650 and the wall of the bores 628b.

The sleeve 594 has a bore 655 formed therethrough. The bore 655 is closed off by an end plug 656 having a bore 657 formed therein which retains a spring 658. The spring 658 engages a spring retainer 660 which acts against the second ball 648 of the first disc 626. The spring retainer 660 and the ball 648 cooperate to assist in axially aligning the force transmitted between the spring retainer 660 and the ball 648. Preferably, the spring 658 has a relatively high spring rate to help stabilize the valve assembly 624 during operation thereof.

A third chamber 662 is generally defined by the sleeve 594 and the wall of the bore 556 of the housing 554. The third chamber 662 is in fluid communication with a high pressure input port, such as the conduit 218 of the brake system 200 of Fig. 11. The third chamber 662 is in fluid communication with the radial bore 644 of the first sleeve 594.

A fourth chamber 666 is generally defined by the end plug 656, the first sleeve 594, the collar 614, the sleeve 600, and the reaction spool 604. The fourth chamber 666 is in fluid communication with a boost output port, such as the supply conduit 216 of the brake system 200 of Fig. 11, via radial passageways 668 formed through the first sleeve 594. Generally, the valve assembly 624 is submersed within the fourth chamber 666.

The operation of the boost valve 550 and the pedal travel simulator 552 shall now be described as being adapted for use in the brake system 200 of Fig. 11. Normally, when the master cylinder 202 is not generating pressurized fluid, the boost valve 550 is in the position illustrated in Figs. 14 and 15 which is similar to the first position 316a, wherein the conduit 328 is in fluid

communication with the supply conduit 216. The ball 630 is unseated from the valve seat 632 of the reaction spool 604, thereby allowing fluid communication between the second fluid chamber 612 to the fourth fluid chamber 666 via the bore 605 of the reaction spool 604. The ball 646 is seated on the valve seat 642 of the first sleeve 594, thereby closing fluid communication between the third and fourth fluid chambers 662 and 666.

When the driver depresses the brake pedal 204, the master cylinder 202 pressurizes the brake fluid in the conduit 210 which flows into the first fluid chamber 572 of the pedal travel simulator 552. The first fluid chamber 572 expands, thereby causing the piston 560 to move leftward, as viewing Fig. 14. The movement of the piston 560 exerts a force on the spring 576 which is transmitted through the first and second spring retainers 578 and 588 to the cap 606 and the reaction spool 604. The force acting on the reaction spool 604 causes the reaction spool 604 to move leftward, as viewing Fig. 14.

The movement of the reaction spool 604 moves the collar 614. As the collar 614 moves leftward, the damping chamber 622 expands. Fluid enters the damping chamber 622 via the orifices 620. The movement of the reaction spool 604 also causes the ball 630 to seat on the valve seat 632, thereby closing fluid communication between the second fluid chamber 612 and the fourth fluid chamber 666. Further movement of the reaction spool 604 pushes against the valve assembly 624 causing the valve assembly 624 to move leftward, as viewing Figs. 14 and 15. Movement of the valve assembly 624 causes the ball 646 to unseat from the valve seat 642 formed on the first sleeve 594. The boost valve 550 is now in a position similar to the third position 316c of the boost valve 310 of Fig. 11. Thus, the pressurized fluid from the fluid pressure generator circuit 214 is allowed to flow into the fourth fluid chamber 666 via the conduit 218, the radial bore 644, and the central bore 640. The pressurized fluid flows around the valve assembly 624 around the second disc 628 and out

through the supply conduit 216. The valve assembly 624 is moved back to the right when the pressure within the fourth fluid chamber 666 rises above the pressure from the master cylinder 12 multiplied by the predetermined boost ratio. The boost valve 550 will eventually reach an equilibrium wherein the balls 630 and 646 are seated on the valve seats 632 and 642, respectively, and will stay in a position similar to the second position 316b of the boost valve 310 of Fig. 11. When the driver changes the pressure generated from the master cylinder 202 by changing the position of the brake pedal 204, the boost valve 550 will operate as described above, thereby maintaining a pressure in the supply conduit 216 at a factor above the pressure generated by the master cylinder 202 as determined by the boost ratio. When the driver releases the brake pedal 204, the piston 560 will return to its at rest position. The boost valve 550 will then return to the position as that illustrated in Figs. 14 and 15.

If the piston 560 is moved the sufficient distance "D", as shown in Fig. 14, the end 569a of the spring 569 will contact the shoulder 570 of the end plug 558. Further movement of the piston 560 causes the spring 569 to compress. Since the shoulder 570 of the end plug 558 is fixed with respect to the housing 554, the spring 569 does not actuate the boost valve 550. The driver of the vehicle feels the reactionary force by compression of both springs 569 and 576. The boost valve 550 is only actuated by the spring 576.

The principle and mode of operation of this invention have been explained and illustrated in its preferred embodiment. However, it must be understood that this invention may be practiced otherwise than as specifically explained and illustrated without departing from its spirit or scope.

What is claimed is:

1. A helical compression spring for use in a pedal simulator of a vehicle brake system, comprising a plurality of coils having varying cross-sectional areas along the length of the spring.

2. The spring of Claim 1, wherein said spring has a progressive non-linear spring rate.

3. The spring of Claim 1, wherein the distance between adjacent coils is generally constant.

4. The spring of Claim 1, wherein the distance between adjacent coils varies along the length of the spring.

5. The spring of Claim 1, wherein said spring has a non-cylindrical inner contour.

6. The spring of Claim 1, wherein said spring has a non-cylindrical outer contour.

7. The spring of Claim 6, wherein said outer contour is conically shaped.

8. The spring of Claim 6, wherein said outer contour is hourglass shaped.

9. The spring of Claim 1, wherein the spring is formed from a wire having a generally rectangular cross-sectional shape.

10. The spring of Claim 1, wherein said spring has a constant helical angle.

11. A spring for use in a pedal simulator of a vehicle brake system, wherein said spring is a coil spring having a plurality of oscillating bends formed in said coil.

12. The spring of Claim 11, said spring having a plurality of coils.

13. The spring of Claim 12, wherein said plurality of coils are positioned in a nested relationship such that radially extending surface areas of said coils facing each other are substantially in contact with one another.

14. A method of manufacturing a spring, comprising the steps of:
a) providing a wire having a generally constant cross-section;
b) forming the wire into a helical compression spring about an axis; and
c) forming a desired axially variant contour on the spring.

15. The method of Claim 14, wherein in step c, the axially variant contour is formed by removing material along an outer contour of the spring.

16. The method of Claim 15, wherein in step c, the outer contour is formed into a conical shape.

17. The method of Claim 15, wherein in step c, the outer contour is formed into an hourglass shape.

18. A pedal simulator for a vehicle braking system, said pedal simulator comprising:

a housing defining a bore therein;

a piston slidably disposed in said bore, said piston cooperating with said housing to define an expansible chamber in said bore on a first side of said piston and a second chamber on a second side of said piston in said bore, said housing defining an opening into said expansible chamber through which pressurized fluids may be directed into said expansible chamber to urge said piston toward said second chamber; and

a spring disposed in said second chamber and acting between said piston and said housing, said spring comprising a plurality of coils having varying cross-sectional areas along the length of the spring.

19. A pedal simulator for a vehicle braking system, said pedal simulator comprising:

a housing defining a bore therein;

a piston slidably disposed in said bore, said piston cooperating with said housing to define an expansible chamber in said bore on a first side of said piston and a second chamber on a second side of said piston in said bore, said housing defining an opening into said expansible chamber through which pressurized fluids may be directed into said expansible chamber to urge said piston toward said second chamber; and

a wave spring disposed in said second chamber and acting between said piston and said housing.

20. A brake system comprising:

a master cylinder for generating pressurized fluid;

a wheel brake in fluid communication with said master cylinder;

a pedal travel simulator having:

a housing having a bore formed therein;

a piston slidably disposed in said bore, said piston and said housing generally defining a fluid chamber being in fluid communication with said master cylinder; and

a spring biasing said piston in a direction to contract said fluid chamber, said spring having a progressive non-linear spring rate;

a source of pressurized fluid; and

a boost valve in fluid communication with said source of pressurized fluid and said wheel brake, said boost valve being actuated by said spring of said pedal travel simulator to supply pressurized fluid from said source of pressurized fluid to said wheel brake via a supply conduit at a pressure ratio greater than fluid pressure generated by said master cylinder.

21. The brake system of Claim 20, wherein said spring is a wave spring.

22. The brake system of Claim 20, wherein said spring includes a plurality of coils having varying cross-sectional areas along the length of the spring.

23. The brake system of Claim 20, wherein said pedal travel simulator includes a second spring acting on said piston and said housing of said pedal travel simulator.

24. The brake system of Claim 23, wherein said second spring is positioned in said housing such that the piston must travel a predetermined distance prior to engagement with said second spring.

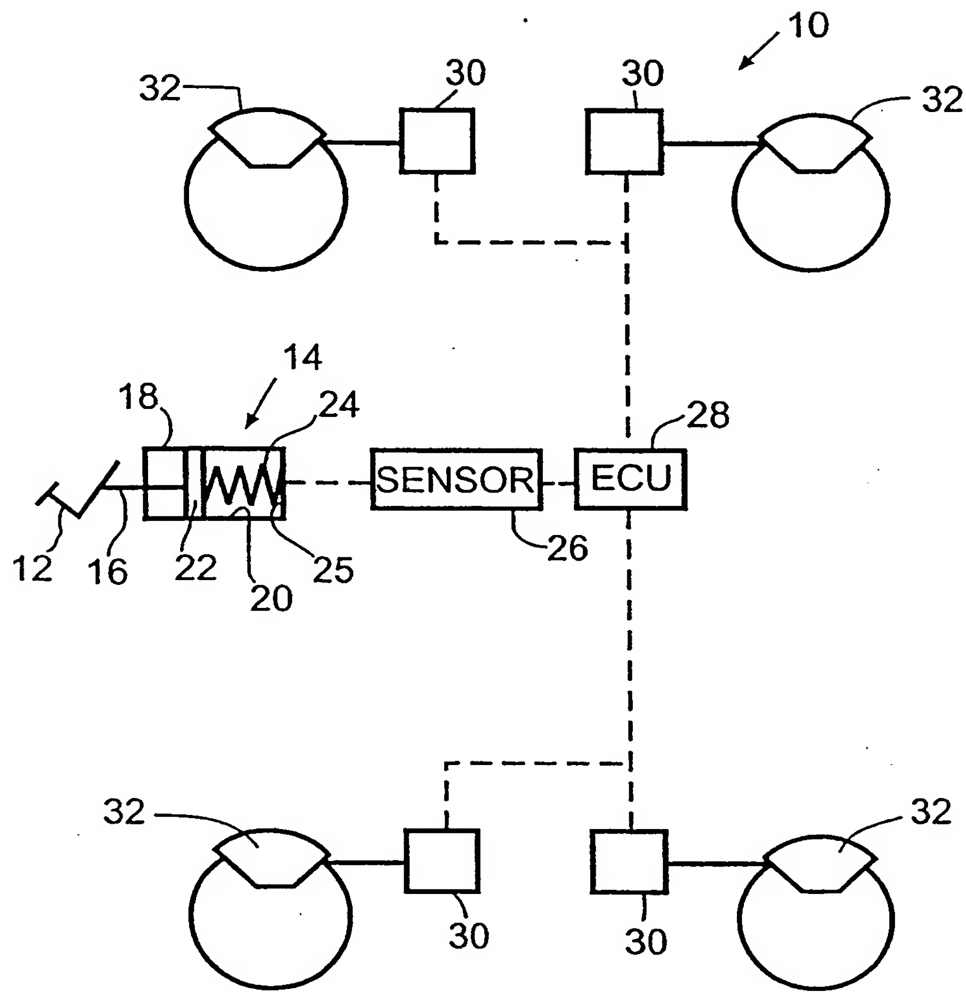

FIG.1

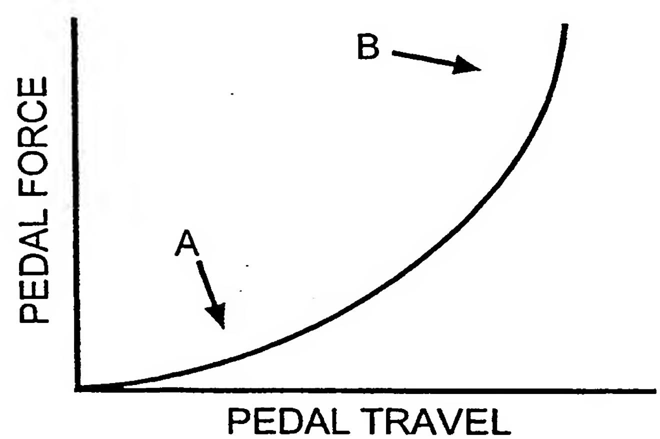

FIG. 2

SUBSTITUTE SHEET (RULE 26)

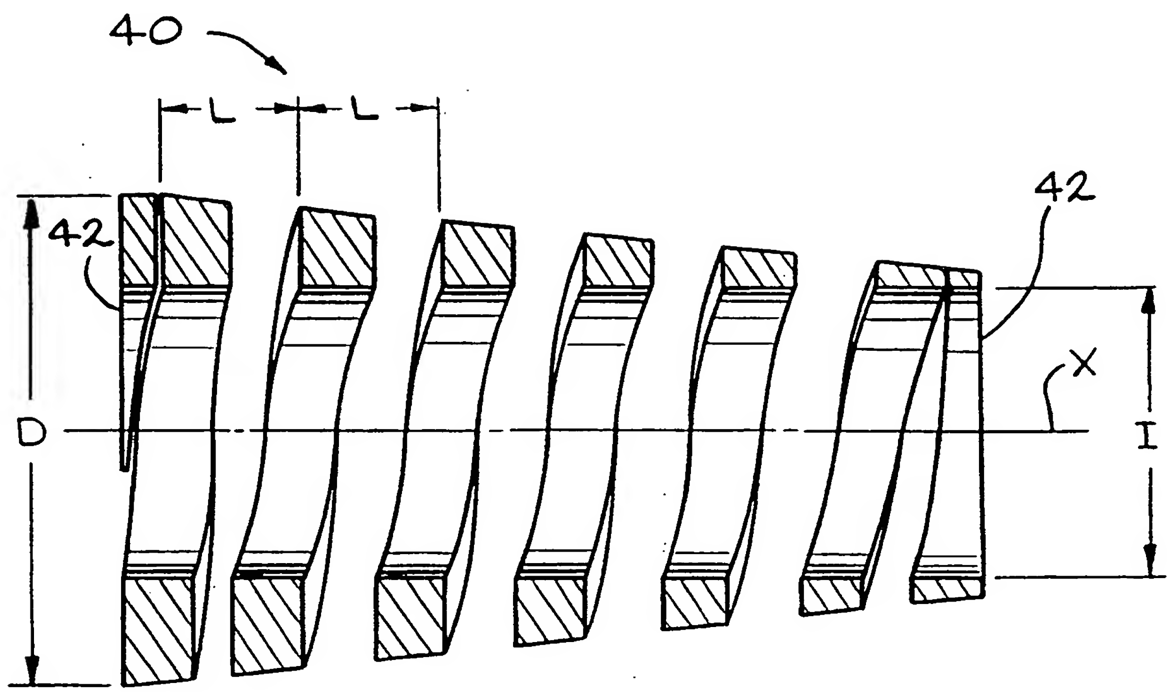

FIG. 3

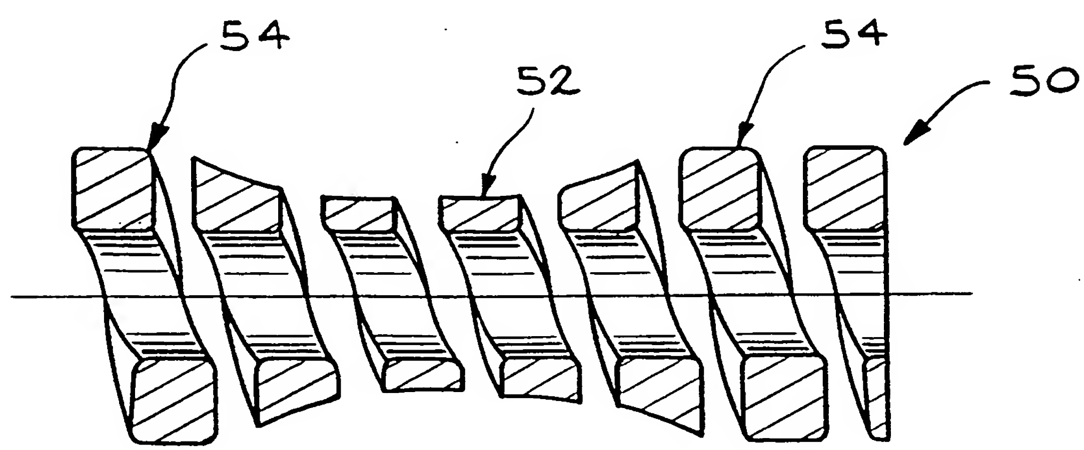

FIG. 4

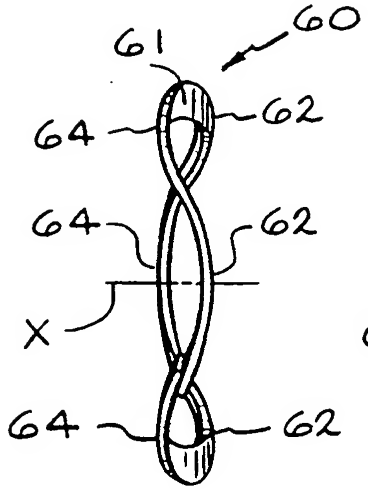

FIG. 5

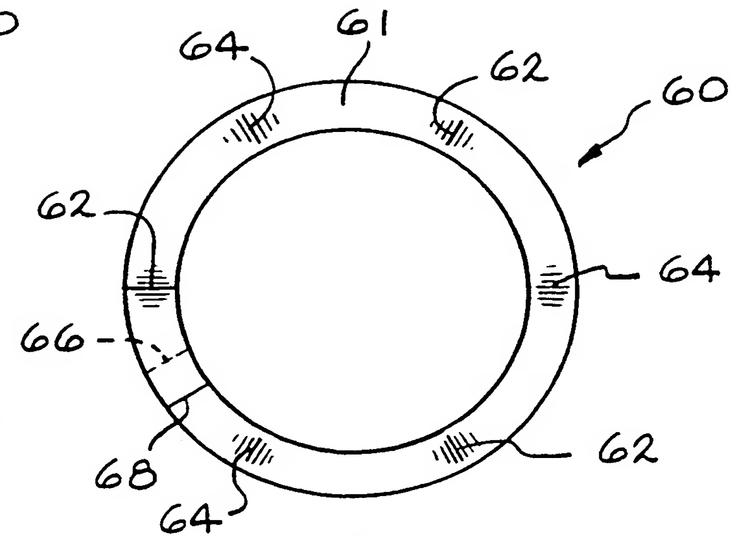

FIG. 6

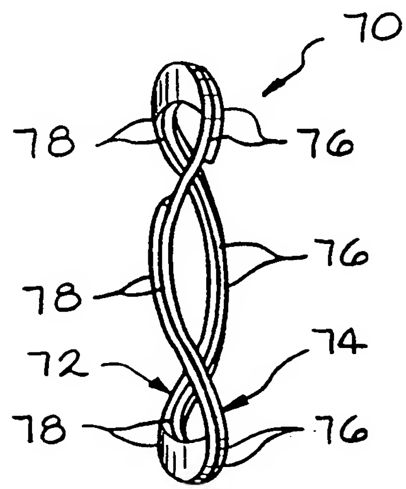

FIG. 7

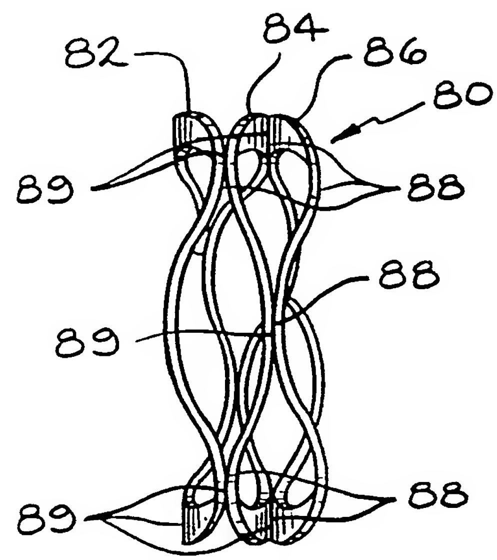

FIG. 8

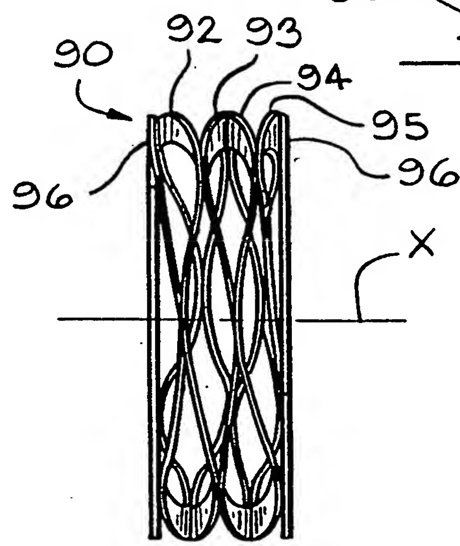

FIG. 9

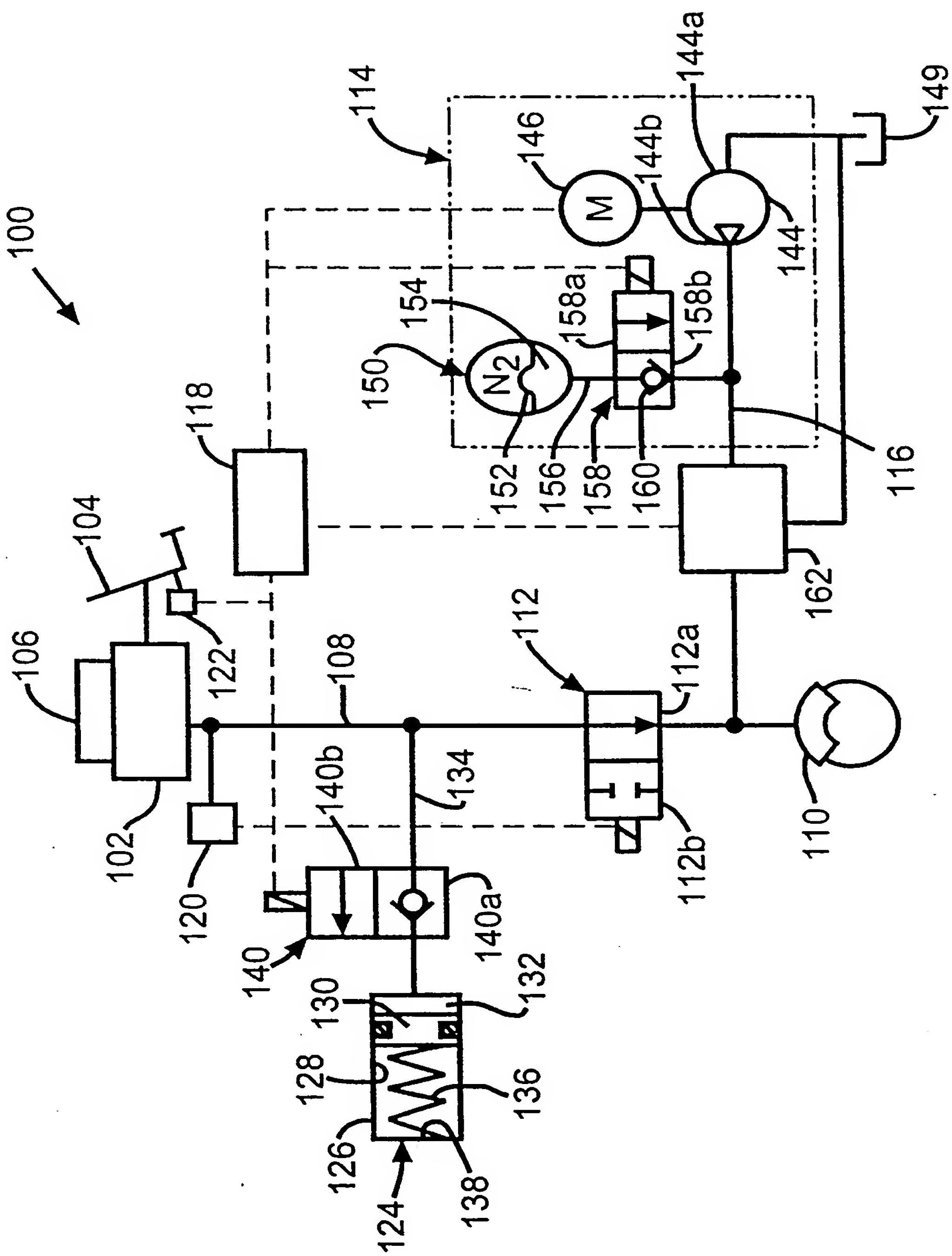
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FIG. 10

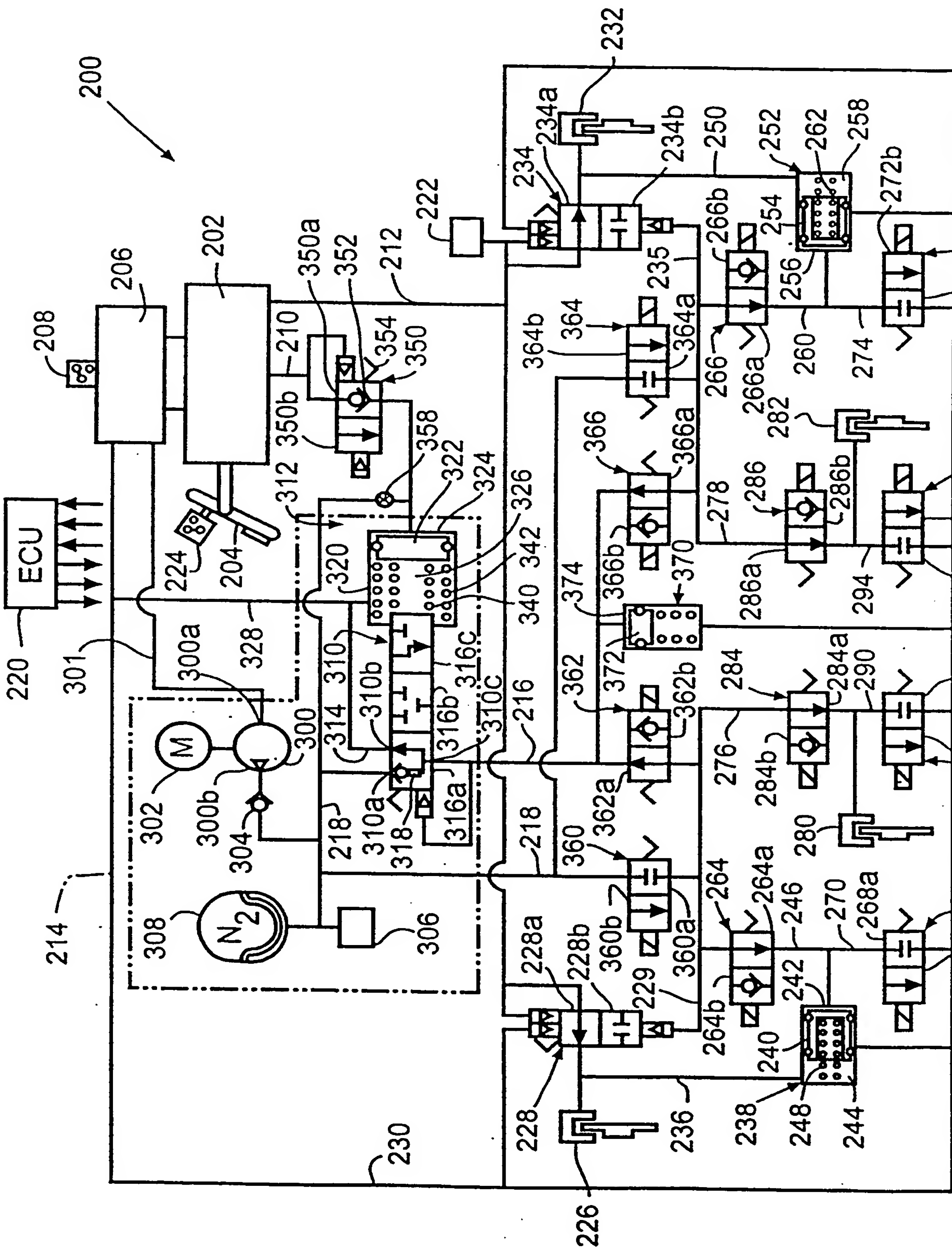

FIG. 11

SUBSTITUTE SHEET (RULE 26)

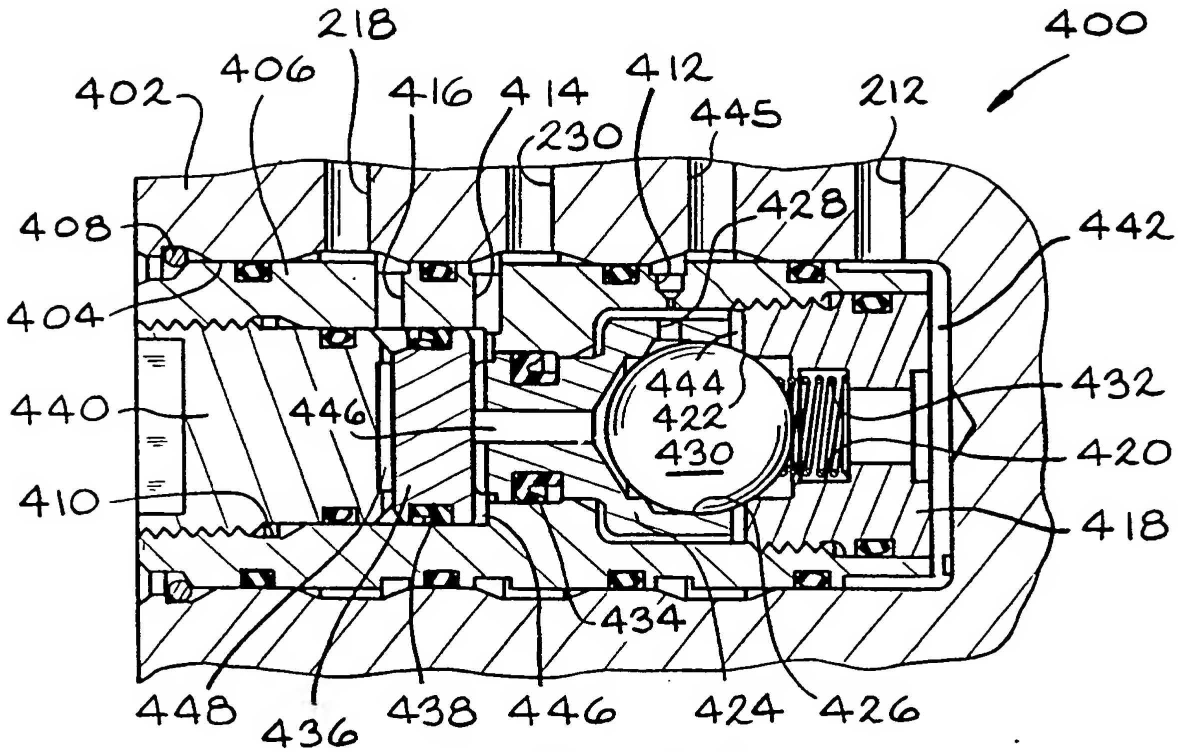
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FIG. 12

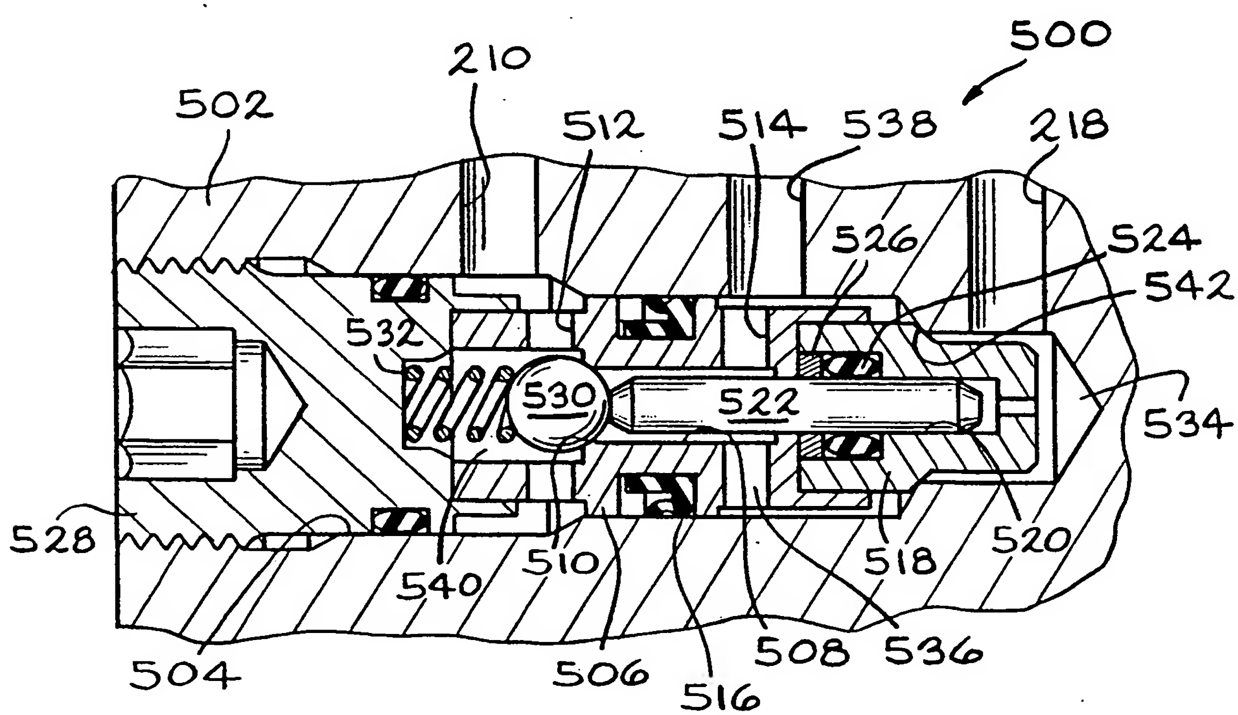
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FIG. 13

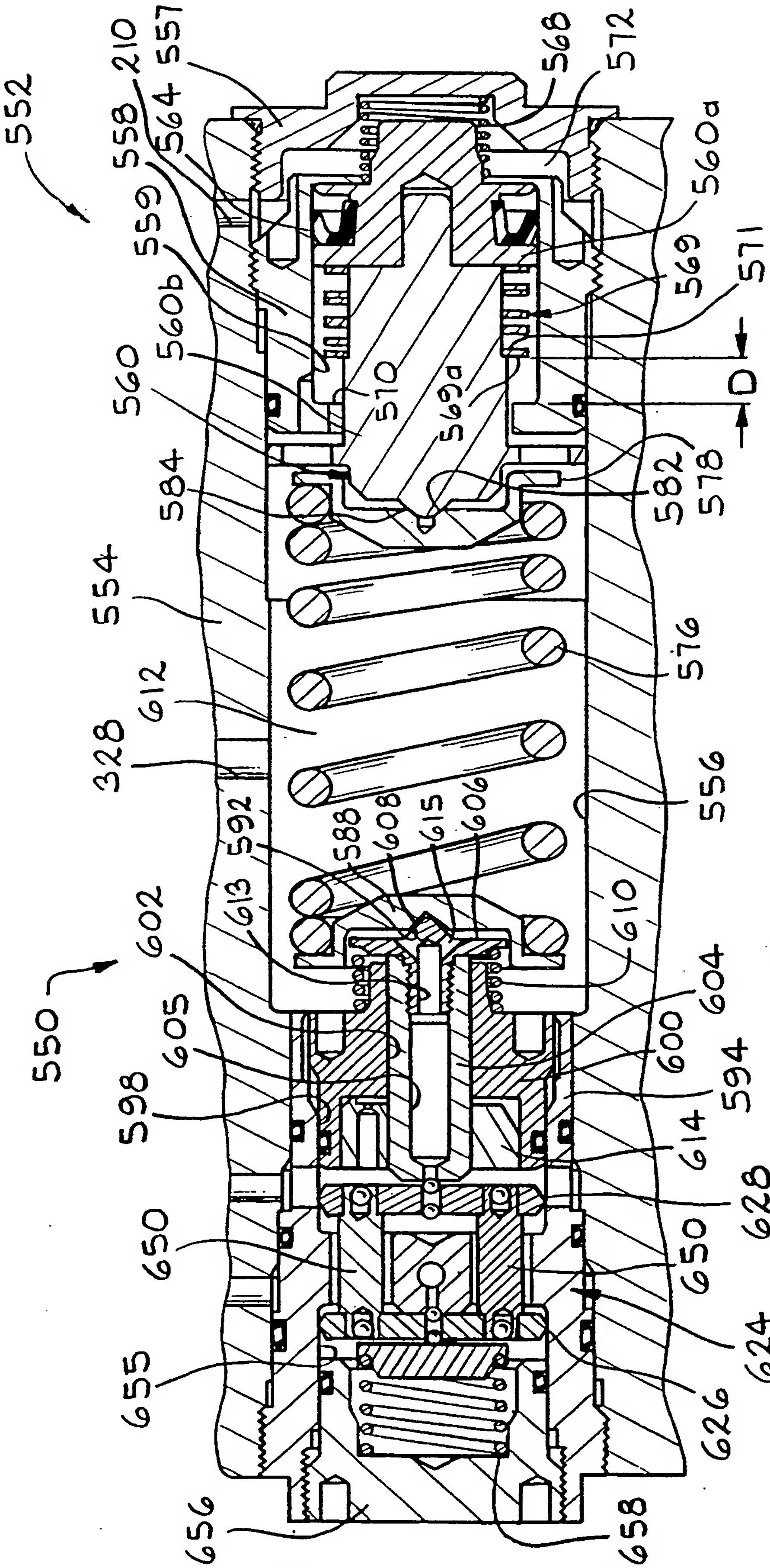
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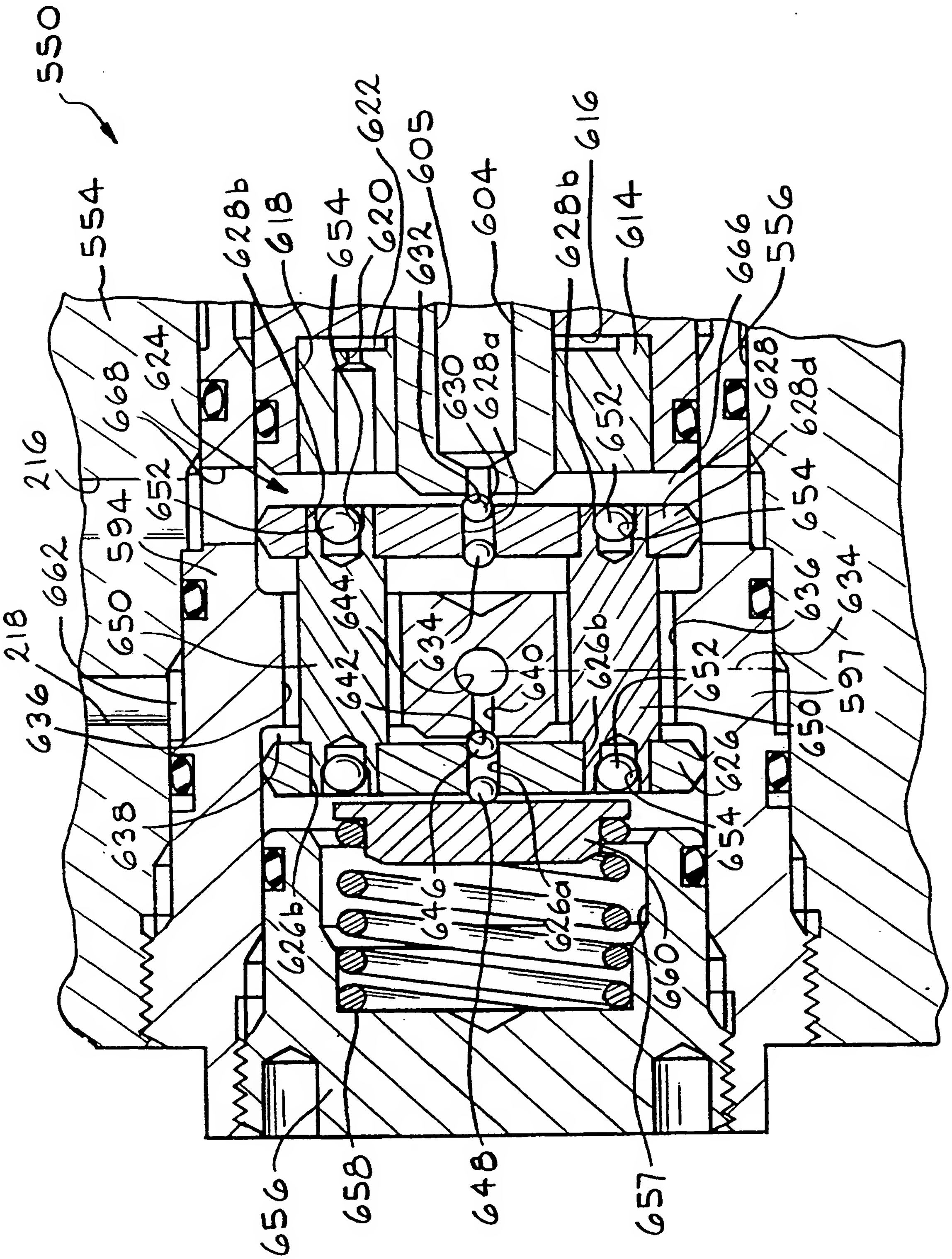
FIG.14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628a
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628d
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FIG. 15

INTERNATIONAL SEARCH REPORT

International Application No
PCT/US 98/26312

A. CLASSIFICATION OF SUBJECT MATTER
IPC 6 B60T7/04 F16F1/04

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 6 B60T F16F

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	FR 508 271 A (CHARLES-LEOPOLD BECK) 6 October 1920	1-3,5-10
Y	see the whole document	17
X	GB 2 241 754 A (SANKO SENZAI KOGYO K.K.) 11 September 1991	11-13
A	see page 4, line 10 - page 5, line 19; figures 1,2	19,20
X	FR 2 090 774 A (DAIMLER-BENZ AG) 14 January 1972	1-3,5, 10,14-16
Y	see page 1, line 26 - page 2, line 27; figures 1-3	17
X	DE 195 46 647 A (ROBERT BOSCH GMBH) 19 June 1997 see column 3, line 26 - column 5, line 40; figures 2,3	20,23

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[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

° Special categories of cited documents :

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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
25 March 1999	01/04/1999

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Harteveld, C

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Internati. Application No
PCT/US 98/26312

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	WO 97 32766 A (KELSEY HAYES COMPANY) 12 September 1997	20
A	see page 3, line 20 - page 8, line 16; figures 1-9	18
X	DE 19 33 385 U (ADOLFO MARTIN VICEN) 8 March 1965 see the whole document	1-7, 14-16
X	DE 85 22 888 U (FICHTEL & SACHS) 10 October 1985 see page 4, line 4 - line 29; figures 2-6	1,2,4,10

1

INTERNATIONAL SEARCH REPORT

Info..ation on patent family members

International Application No
PCT/US 98/26312

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
FR 508271 A	06-10-1920	NONE	
GB 2241754 A	11-09-1991	DE 4008509 A	19-09-1991
		DE 9007634 U	17-06-1993
		FR 2660035 A	27-09-1991
FR 2090774 A	14-01-1972	DE 2020678 A	25-11-1971
		GB 1323187 A	11-07-1973
		US 3727902 A	17-04-1973
DE 19546647 A	19-06-1997	WO 9721573 A	19-06-1997
		EP 0863826 A	16-09-1998
WO 9732766 A	12-09-1997	AU 2318997 A	22-09-1997
		DE 19781643 T	25-03-1999
		GB 2327728 A	03-02-1999
DE 1933385 U		NONE	
DE 8522888 U	10-10-1985	FR 2602839 A	19-02-1988
		GB 2179424 A,B	04-03-1987
		US 4810231 A	07-03-1989